BY THE SAME AUTHOR

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MARINE AND STATIONARY
DIESEL ENGINES

DESCRIBED AND ILLUSTRATED
WITH NUMEROUS

ORIGINAL FORMULÆ

FOR THEIR

DESIGN

AND

INSTRUCTIONS FOR INSTALLATION
AND

OPERATION

BY

A. H. GOLDINGHAM

Member of the Institution of Mechanical Engineers; Member of the American Society of Mechanical Engineers; Member of the Society for the Promotion of Engineering Education

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SECOND EDITION REVISED AND ENLARGED
WITH

APPENDIX

(Diesel Engine Castings by F. J. COOK)

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PREFACE

This is essentially a practical treatise. The Diesel engine, both stationary and marine, having attained such prominence all over the world, the want has been felt for a practical work in concise form, giving all information available regarding the various types and designs of this engine.

To supply this need, this treatise is offered. It will be found of value to the consulting engineer, to the draughtsman, to those who are investigating the subject, as well as to those who propose to install engines.

All existing literature pertaining to this subject has been consulted. The formulæ for design and construction have each been proved and carefully checked, many of which are original.

The chapter on "Operation and Correction," which has been compiled with great care, should be of distinct value to those who wish to study its operation or who are operating Diesel engines.

The discussion on marine engines, and the technical description of the many different types of horizontal, vertical, slow speed and high speed, marine and stationary engines, drawing attention to special features of construction of these types, should be of value both to the technical and non-technical reader.

This work is copiously illustrated. The writer has been fortunate in having placed at his disposal draw-

ings, photographs and sectional views of nearly all of the leading Diesel engines made.

The writer is indebted to the following publishers and manufacturers for their courtesy in placing illustrations and other matter at his disposal for reproduction:

The publishers of Engineering, London, for permission to reproduce the illustrations of the Diesel engines in the ship "France."

The publishers of Cassier's Magazine, London, for permission to reproduce illustrations of various sprayers.

The American Society of Mechanical Engineers, for extracts from the address of the late Dr. Rudolf Diesel.

Messrs. Burmeister and Wain, Copenhagen, who kindly furnished the illustrations of their engine.

Messrs. McIntosh and Seymour, Auburn, N. Y., who supplied the illustrations of their engine.

The U. S. representative of the Werkspoor Company, for furnishing the illustrations of that engine.

The Rudder Publishing Company, for furnishing other illustrations.

The New London Ship and Engine Company, Busch-Sulzer Bros. Diesel Company, and the Snow Pump Company have each furnished information regarding their respective engines.

The publishers of "Internal Combustion Engineering," London, for their kind permission to make ex-

tracts and reproductions from their paper of the descriptive matter of some of the various engines referred to.

Mr. R. C. Crowly has made valuable suggestions and has kindly corrected some of the proofs.

The assistance and courtesy extended by the above and all others who have assisted in the preparation of this treatise is hereby acknowledged.

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CHAPTER I

INTRODUCTORY

The Diesel engine, the subject of this treatise, was the invention of the late Dr. Rudolf Diesel. It was described by him in his original patent specification dated August 27, 1892. The first commercial and successful engine was constructed and completed in 1897 at the Augsburg Works, Augsburg, Germany. It attained the highest thermal efficiency in comparison with all other heat motors then designed, and still at this date and state of knowledge, while other types of internal combustion engines may claim equal results and some advantages, in operation as compared with it, still, no other process is known by which a higher efficiency is obtained of changing heat into work. A thermal efficiency of 48 per cent or an effective efficiency of nearly 35 per cent was claimed by the inventor for it in some cases. Dr. Diesel in his treatise, the Rational Heat Motor, described the unique features of the working cycle of his engine as follows:*

"The characteristics of the working cycle, as deduced from our theory of combustion, are:

"1. Production of the highest temperature of the cycle, not by and during combustion, but before and independently of it, solely by compression of air, or of a mixture of air and gas.

*"Theory and Construction of a Rational Heat Motor," published by E. & F. N. Spon, London, 1894.

"2. Gradual introduction of the pulverised combustible into the compressed and heated air during part of the stroke of the piston, and in such a way that combustion produces no increase in temperature of the gases. Hence the combustion curve obtained is as nearly as possible isothermal. After ignition, combustion is not left to itself, but is regulated by external mechanism throughout its course, and the right proportions are thus established between the pressures, volumes and temperatures of the gases.

"3. Choice of the right weight of air in proportion to the heat value of the combustible, the temperature of compression (and of combustion) being previously so determined that the working of the engine, lubrication, etc., can be carried out without cooling the cylinder.

"We need scarcely say that, in practice, these conditions can hardly be realised. In working engines the corners of the indicator diagrams are more or less rounded, the curve of combustion is not perfectly isothermal, and under given conditions a certain increase in temperature or pressure will be caused by combustion. The rise, however, is slight in comparison with that previously produced in the temperature and pressure of the gases by compression. Nor do these differences affect the principle of the cycle."

The theory of the Internal Combustion Engine is so fully treated in many text-books on this subject and the Diesel Engine, its theory and cycle of operation has been so widely discussed. and so fully written

about and compared with other engines and cycles, that only a brief reference will here be made, outlining the chief features of that cycle and noting generally the difference between it and other internal combustion engines.

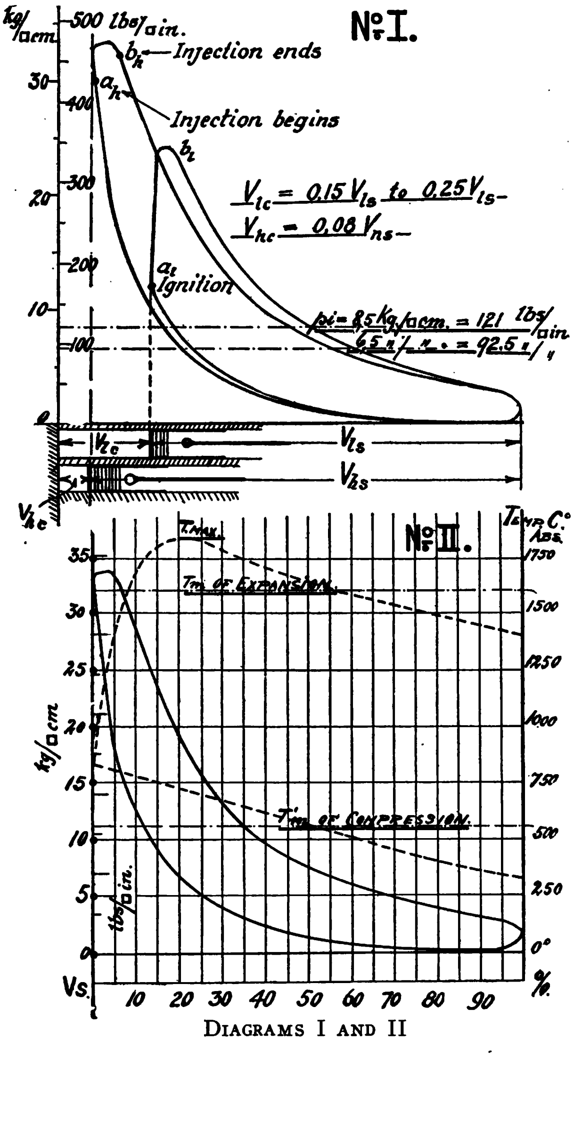
All Internal Combustion engines work on either of the following theoretical thermodynamic cycles:

(a) Combustion at constant volume,
(b) Combustion at constant pressure

The first named type is exemplified in the Beau de Rochas, "Otto" or 4 cycle gas or oil engine, where the total heat is evolved while the volume of gas remains practically constant, and the rejection of heat takes place under the same conditions.

The second type is that engine in which all the heat is taken in while the pressure remains constant in the cylinder and is rejected under similar conditions. The Diesel engine is the example which most nearly reaches the theoretical condition of this cycle.

DIAGRAM No. I—This consists of two diagrams superimposed and serves to illustrate the comparative pressures and other particulars of these two types of engines. The pressures in kilograms per square centimeter and lbs. per sq. in. are both given. In the lower part of this diagram are shown the relative clearances of each type and the mean effective pressures (pi) are indicated. The smaller diagram represents the operation of the constant volume engine where the com-



DIAGRAMS I AND II

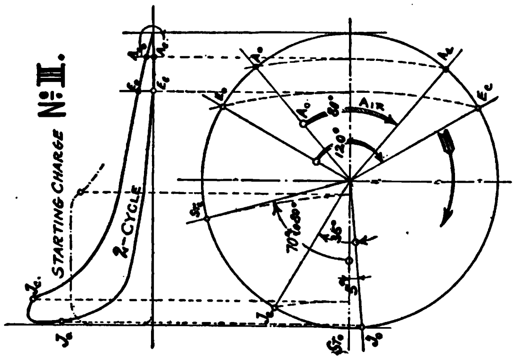
bustible charge of air and fuel is burned suddenly at approximately constant volume.

The larger diagram represents the operation of the Diesel Engine where a much higher range of pressures obtains in the cylinder and where combustion takes place at practically constant pressure. This is shown by the ignition line at the top of the diagram which is almost horizontal. The points in the cycle where fuel injection begins and ends are each indicated, likewise the point of ignition on the low pressure diagram is given.

The clearance of the low pressure engine is between 15% and 25%, while that of the Diesel 4-cycle type is approx. 8%. It will also be noted that the M. E. P. of the Diesel type is greater than the other, and the maximum pressures being so much greater the resulting load and strains on the piston, reciprocating, and revolving parts, has to be allowed for. The compression lines which should coincide, for the sake of clearness are made to slightly diverge.

DIAGRAM No. II shows the relative temperatures and pressures of a Diesel engine at the various positions of the piston during the compression and expansion strokes. The temperatures shown are the absolute temperatures as computed by Mallard & Le Chatelier, based on the change of specific heat.*

*For an exhaustive treatment of this subject the reader is referred to Güldner's well-known treatise on internal combustion engines, "The Design and Construction of Internal Combustion Engines."



E_o Exhaust opens
E_c " closes
A_o Airvalve opens
A_c " closes
J_o Injection begins
J_c " ends
St_o Startingvalve opens
St_c " closes

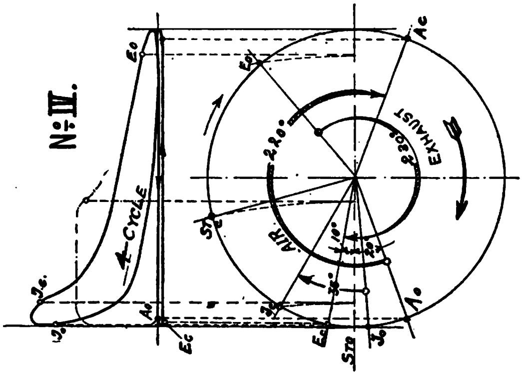
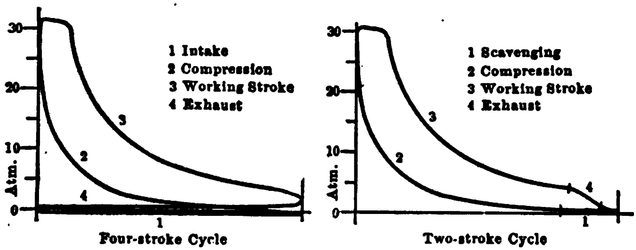


DIAGRAM III AND IV

It is of interest to note that the highest temperature exists at about 20% after the piston has passed the inner dead center, being also 7% after the close of the fuel and air injection valve. Mean temperatures of both compression and expansion strokes are also indicated.

CYCLES. All gas or oil engines operate on either the 4 cycle or 2 cycle (sometimes referred to as the 4 stroke cycle or Otto cycle and also as the 2 stroke cycle), which means that either 4 strokes of the piston are re-



DIARGAM V

quired to complete the cycle of operation, which cycle is repeated indefinitely or that the engine completes the same process in two strokes of the piston. The operation of the 2 and 4 cycle types of engines, is shown in comparison diagramatically together with the indicator cards for each type in Diagrams III, IV and V.

THE TIMING OF THE VALVES, especially the fuel inlet valve, will be found to vary somewhat with the differ-

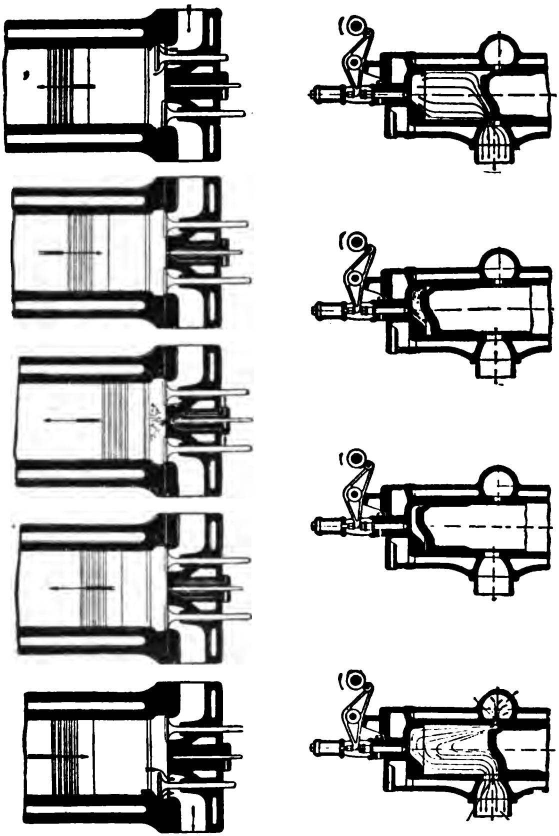
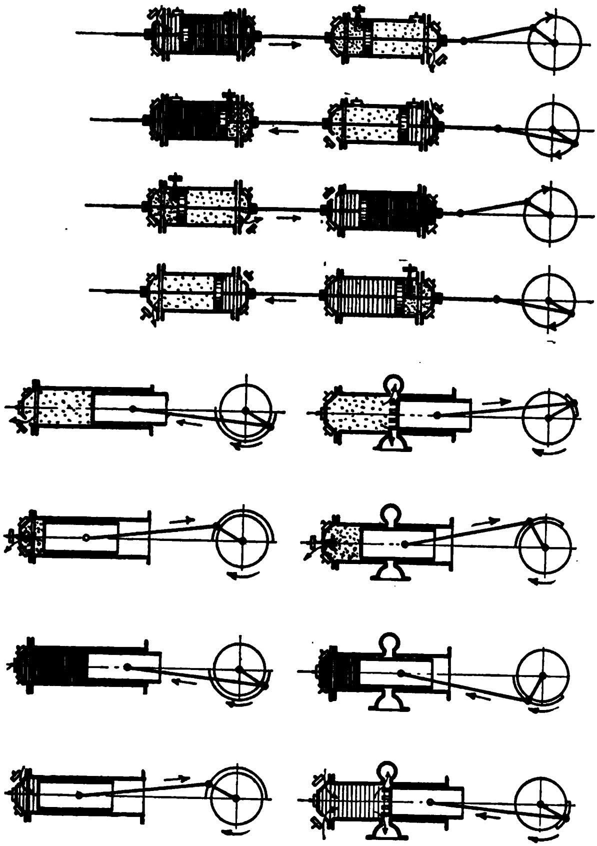


Diagram VI



Compression of pure air.

Inlet of pure air.

Combustion and expansion.

Exhausting of burnt gases.

DIAGRAM VII

ent speeds of engines. The periods shown indicate average conditions.

Diagram No. VI shows the actual position of the piston in the cylinder of both 2 cycle and 4 cycle types at each period of their cycles. The position of the air and exhaust valves during each stroke of the 4 cycle is shown, likewise the air inlet, and exhaust of the 2 cycle and the period of fuel injection in both are each indicated.

Diagram VII shows the different processes of operation of the different types of two and four cycle engines.

Diagram VIII shows again the periods of valve motion, the two cycle in the upper diagram and the four cycle below it.

In all gas or oil engines previous to 1889 the air necessary for combustion and the fuel from which the heat was to be evolved were introduced into the cylinder or combustion space before compression was completed, thus in all such engines the degrees of compression allowable was regulated by the temperature of ignition of the mixture of fuel and air in the cylinder.

Compression of the mixture above the pressure at which the temperature of ignition of the mixture was reached, would have involved preignition (that is, ignition before the piston reached the dead center), with consequent loss of power and other unsatisfactory results.

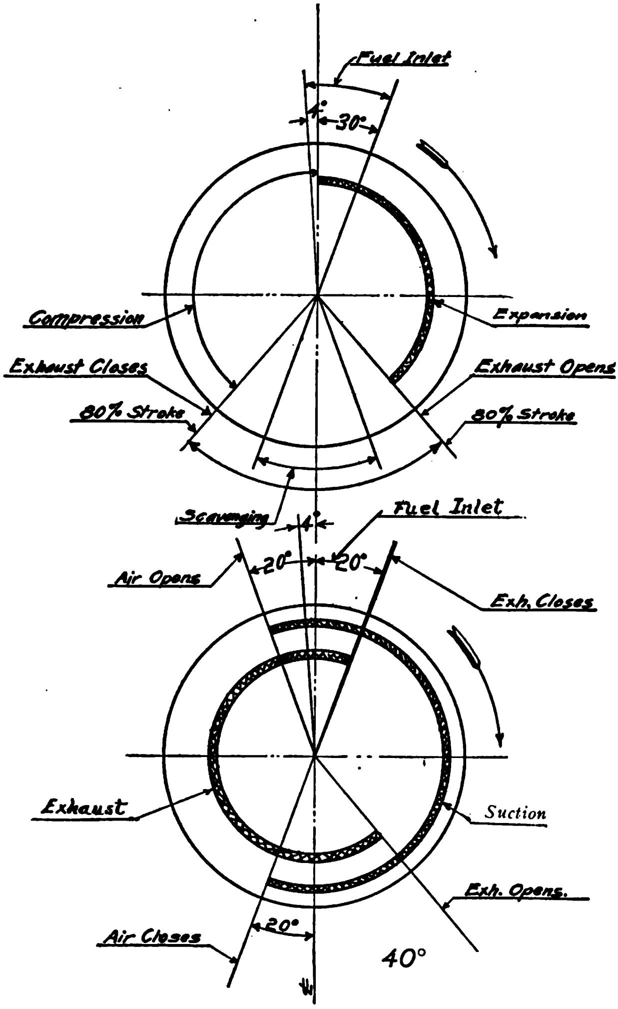


DIAGRAM VIII

The Diesel cycle of operations (first described and published about 1892) differs from those previously named in two essential features:*

1st—Ignition and combustion is caused by very high compression, and

2nd—Air only is taken into the cylinder and thus, during the compression period air only is in the cylinder; after the compression of the air is completed, then the fuel is introduced into the cylinder, which fuel in most types of Diesel engines is properly mingled with high pressure air, which thoroughly atomizes the fuel. Injection of the fuel into the combustion space and its ignition are simultaneous because the highly compressed air in the cylinder has been heated by compression to a temperature always sufficient to cause the ignition and combustion of the fuel which

*The conception by its inventor of the Diesel engine was originally along the following lines, which were subsequently modified, as can be readily understood by comparing the proposed conditions outlined in the following remarks with the Diesel engine actually developed:

1. In order to insure a high temperature, it would be necessary to produce sufficient pressure by the compression of pure air in the cylinder prior to the injection of the fuel. This compression should be isothermal in its first stage, the heat of compression being absorbed by the injection of water, but should be completed adiabatically.

2. The injection of the fuel should be so regulated in its ratio, as to attain conditions of constant temperature during combustion. The fuel should be injected in so finely divided a state that its combustion or burning should result without shock immediately it comes in contact with the heated air.

3. The combustion at constant pressure should be followed by an adiabatic expansion of the heated gases down to atmospheric pressure.

takes place at constant pressure. The increasing volume of the burning mixture, as will be seen from the indicator card, maintains the pressure in the cylinder as the piston commences to move outwards.

Theoretically the Otto Cycle is more efficient than the Diesel Cycle, as with the same initial temperature and pressure the Otto Cycle Engine would have a higher ratio of expansion and consequently a greater temperature drop shown in the expansion curve than with the Diesel engine. In practice however the conditions are reversed, as it is not yet possible with the Otto Cycle Engine to compress the explosive mixture sufficiently high to obtain an initial temperature of combustion equal to that secured in the Diesel Engine.

THE INCREASED EFFICIENCY of the Diesel Engine over the other types is due largely to the following causes:

(a) Greater difference between initial and final pressure of expansion.

(b) Slight increase of efficiency by decreased loss of heat to the exhaust gases and cooling water jacket. At the period when combustion commences a minimum amount of cylinder wall surface is exposed, due to the higher compression pressure.

(c) Complete combustion of the fuel. As the temperature of the compressed air in the cylinder at about 32 atmospheres is much greater than is actually necessary to ignite the incoming charge of fuel and air, combustion is, therefore, rapid and complete and owing

to the sustained high temperature during combustion in the cylinder continues till completed.

(d) Complete pulverization or atomization of fuel before entering the cylinder—proper mingling of the minute particles of fuel with oxygen in the air.

THE THERMAL EFFICIENCY of the modern Diesel 4 cycle engine operating at full load capacity, taking the actual or Brake H.P., reaches 32% to 35%, or with the indicated H.P., 41% to 47%. The 2 cycle engine under similar conditions is approximately 3% less than the 4 cycle. Taking the temperature of compression as 1000° C. or 1273° Abs. and the temperature of the exhaust as 300° C. or 573° Abs., theoretically the maximum thermal efficiency possible would be equivalent to 55%.

THE HEAT BALANCE of the average Diesel engine showing ordinarily the disposition of the heat received into its cylinders from the fuel is as follows:

TABLE I

	Per cent.	B. T. U.
Heat equivalent of work done plus friction and pumps..........	43.	7955
Carried away by cooling water....	33.	6105
Rejected in exhaust gases........	23.	4255
Radiation and other losses........	1.	185
	100.	18,500

CHAPTER II

DETAILS OF CONSTRUCTION

THE DIESEL ENGINE is now manufactured in various types both of the horizontal and vertical design. By far, the larger number of makers are building vertical engines. Not only for marine work but stationary engines are also more favored of the vertical type, while some prominent makers are building horizontal engines.

Most builders are manufacturing single acting rather than double acting type engines.

For the information of those who are not familiar with the design and construction of the Diesel Engine, the following information is offered. It is hoped it will be found of value both by the designer, the engineer and non technical reader.

The construction of the chief parts of a representative Diesel Engine are shown, the strains each is subjected to, and the strength requisite in such parts to withstand these strains and temperatures.

THE FOLLOWING FORMULÆ have been compiled mostly from actual experience, all publications on the subject* have been consulted and the formulæ given have been carefully checked. Some of the formulæ

*For a most comprehensive treatment of the subject of design of Diesel Engines the reader is referred to Güldner's "The Design and Construction of Internal Combustion Engines" or "Die Gasmotoren," by Hermann Haeder.

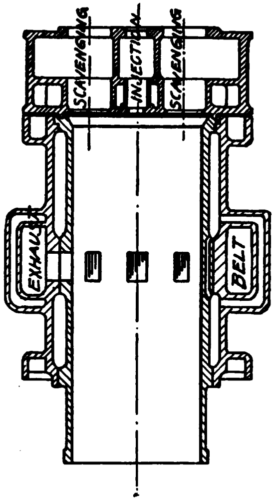


Fig. 1

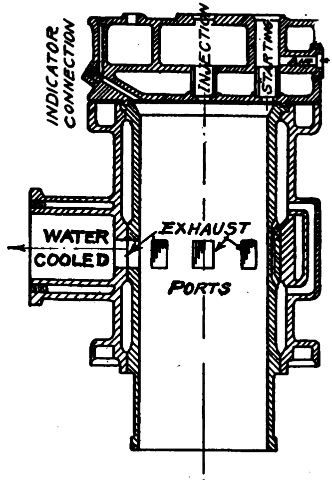


Fig. 2

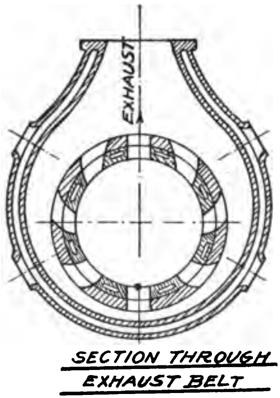


Fig. 3

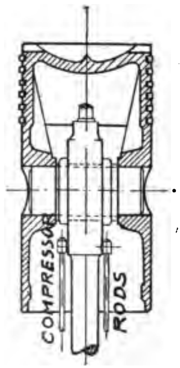


Fig. 4

are in a simplified form or somewhat modified from formulæ in other publications.

CYLINDER—Figs. 1 and 2 show in section a 2 cycle type cylinder and cylinder head or cover. Fig. 3 shows sectional view of exhaust belt. The material for cylinder liner whether it is made separately from the casing, or integral with it, should be hard close grained iron having a tensile strength of about 32000 to 35000 lbs. per sq. in.

All internal parts of Diesel engines have to withstand not only the high pressures but also high temperatures which are shown approximately in diagram 2, which temperatures under certain conditions may temporarily considerably exceed those shown.

The large amount of heat transmission to which the walls of an internal combustion engine are subjected was pointed out by Prof. Junkers in his investigations of this subject.* He found that through the walls of the combustion chamber the heat transmitted is about nine times as great as that transmitted in a steam boiler firebox during a stated period and area.

The combustion pressure† p varies from 460 to 600 lbs. per sq. in. For high speed engines and also where heavy tar oils are used, pressures of 700 lbs. per sq. in. are developed. Assuming an average pressure of 510 lbs. per sq. in. of piston area, the pressure upon the piston (P) would be:

*Investigations and Experimental researches for construction of large oil engines, H. Junkers, Berlin, Nov., 1911.

†Guldner takes combustion pressure $p = 500$ lbs.

$$P = 510 \times 0.785D^2 = 400D^2$$

where D = Diameter of cylinder.

The thickness (S) of cylinder liner may be taken as:

$$S = 0.07D \text{ inches.}$$

To this dimension of the thickness of metal should be added ¼″ according to the diam. of the cylinder, which allowance is made for reboring the cylinder when it becomes worn. Cylinders having more than 15″ diameter may have the thickness of metal of the liner gradually decreased toward the open end.

The thickness there (S_1) may be taken as follows:

$$S_1 = \tfrac{3}{4}S.$$

These dimensions, however, may be slightly decreased in the large cylinders and increased in the smaller sizes.

In order that the cooling effect of the water circulating around the cylinder may be as efficient as possible, the cylinder liner should not be made heavier than necessary.

THE CYLINDER JACKET WALL in the direction of its axis is under a pulling force of $P = 400\, d^2$. The cross sectional area $a = \pi\, D\, s$. The stress per sq. in.

$$p = \frac{400 \times d^2}{\pi\, D\, s} \text{ or } s = \frac{400 \times d^2}{\pi\, D\, p}$$

and let p = 1800 lbs. per sq. in.

we get $s = 0.071\, \dfrac{d^2}{D}$.

where $D =$ mean diameter of cylinder jacket wall.
$s =$ thickness of " " "
$d =$ diam. of piston.

EXHAUST VALVES are of such size as to allow a mean velocity of gases through them between 75 and 120 ft. per second.

$$a = \frac{V}{v} \text{ sq. ft.}$$

Where $V =$ displacement of cyl. in cu. ft. per second
$v =$ velocity in ft. per second.
$a =$ area of valve in sq. ft.

An alternative formula is:

$$a = \frac{F \times c}{v} \text{ sq. ft.:}$$

where:

$c =$ piston speed in ft. per second.
$F =$ area of piston in sq. ft.

EXHAUST PORTS with two cycle engines occupy half the circumference of the cylinder and are of a height to allow proper opening and closing by the piston.

CYLINDER HEAD BOLTS. The material for cylinder head bolts is good wrought iron or soft steel with an allowable tensile stress of 5500 to 6500 lbs. per sq. in.

Size of bolts is determined by the maximum pressure $P_t = 400D^2 + 20\%$ to 30% added for tightening. In some designs numerous small diam. bolts are em-

ployed, in others, where they pass through the head, larger bolts and fewer in number are used. The bolts necessary for the connection of the cylinder to engine frame are to be similarly computed.

CYLINDER COVER: For the cylinder cover or head, a part of Diesel engines which, in the past, has given more trouble than probably any other part of the engine, a soft charcoal iron is recommended, tough and close grained.

Thickness of metal (S) of cylinder cover is difficult to compute because of complexity of form and unknown casting strains. A simple design has:

$$S = \sqrt{\frac{r^2}{6}}$$

where

S = thickness of metal of the inner wall of cover.

r = radius of the largest circle it is possible to describe on the plain surface of metal existing between the different openings for valves, etc., as shown in sketch, Fig. 5.

GASKETS—Copper wire gaskets are recommended for joints between combustion chamber and valve housings as well as air compressor joints. Between cylinder head and cylinder liner a copper ring or graphited Klingerit gasket is used.

THE PISTON uncooled is shown at Fig. 4 and the water cooled or oil cooled piston and crosshead is

shown at Fig. 6. This part is also made of hard close grained cast iron, preferably not so hard as cylinder liner, length of piston (L) is approximately 2.3 diameter where trunk piston and no crosshead is used. The

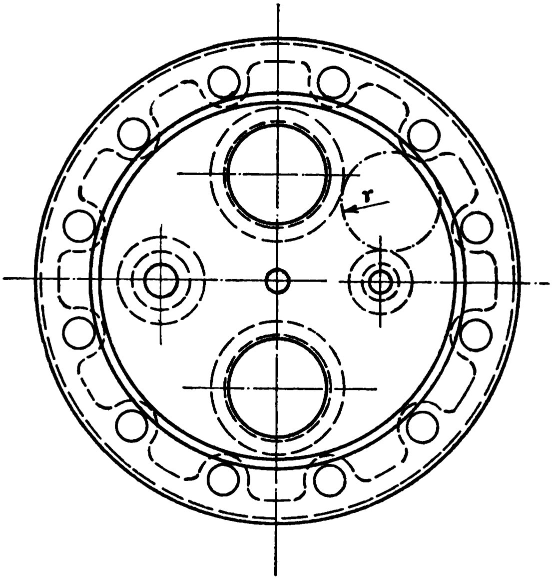


FIG. 5

piston when used in a 4 cycle type having a crosshead and guides is of length only necessary to contain properly the piston rings. With the 2 cycle double acting type again, the length of the piston is governed by the proper opening and closing of the ports, and with the 2 cycle single acting type the length of the piston is

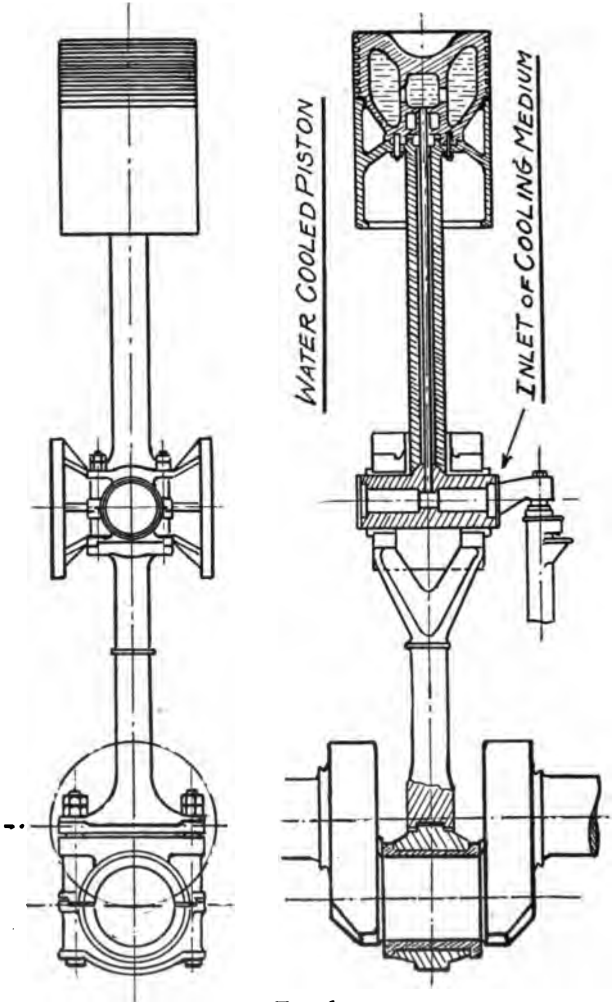


FIG. 6

such that the exhaust ports remain covered by it when at the inner dead centre.

The maximum pressure of piston due to the lateral thrust of connecting rod should not exceed 20-30 lbs. per sq. in. of its projected surface, taking the piston and also including the piston Ring space. Thickness of end of piston (S)

where R = radius of piston:

$$S = 0.23\,R \text{ to } 0.27\,R.$$

The thickness of metal being less where piston is not flat, but is cone shaped at its end. If the piston end is supported by ribs or otherwise besides at its circumference, then its thickness can be decreased.

Thickness of metal of piston barrel is:

$$S = \frac{D}{30} + t + 0.4.$$

Where t = depth of piston groove.

In cases when a piston rod is used and thus the strain on this metal is almost eliminated, then this dimension can be decreased.

Thickness of open end of piston may be

$$\frac{S}{2} \text{ to } \frac{S}{3}$$

The part of piston exposed to heat is slightly reduced in diameter to allow for expansion. This reduction may be 3 thousandth inch per inch diam. of piston.

CROSSHEAD: Engines with 150 B. H. P. in one cylinder or of greater power a crosshead and guides are preferred. The different advantages of the crosshead are:

1. The guides within which it works are maintained at an even temperature and are not subject to expansion and contraction of the cylinder which affect the trunk piston.

2. Lubrication. It is simpler to lubricate the crosshead which does not come in contact wth the heated parts of the engine as does the trunk piston.

3. Adjustment. As the guides of the crosshead become worn they can be easily adjusted, whereas the trunk piston does not allow of adjustment for wear.

4. Piston Seizing. The possibility of the piston seizing through overheating or improper lubrication is minimized when the crosshead is used.

The above remarks refer to the single-acting engines—with the double-acting type, of course the crosshead is always necessary.

WRISTPIN: This should be the best Machinery-steel; surface hardened. Bearing Pressure on it should not exceed 1800 lbs. per sq. in., projected area. The diameter of wristpin (d) is as follows:

$$d = \sqrt[3]{\frac{D^2 \times L}{12}}$$

$$l = \frac{400 D^2}{d \times 1800} = \frac{0.25 D^2}{d}$$

Where $L =$ distance between center of lugs of piston holding wristpin.
$l =$ length of bearing surface of pin.

PISTON RINGS:

$s =$ thickness of Ring.
$b =$ width of Ring.
$c =$ space between grooves.
$b = 1\frac{1}{4}$ to $2.\ s. \quad c = > b.$

PISTON RINGS are made of different dimensions by different builders. A piston ring narrow in width and comparatively of greater thickness which has the advantage of maintaining its shape when exposed to the higher temperatures, is used by some builders and then the following dimensions are used.

$s = 0.5\ b$ on larger engines.
$s = b$ on smaller engines.
$b =$ usually $\frac{3}{8}''$.
Number of piston Rings $= 6$.

CONNECTING ROD BOLTS:

These are made of open hearth Steel or tough Norway Iron. They are under a load due to the inertia of the reciprocating parts, also to shocks resulting from worn bearings of connecting Rod.

THE WEIGHT OF THE RECIPROCATING PARTS is 10 to 15 lbs. per sq. in. of piston, which makes the inertia load amount to about 90 to 100 lbs. per sq. in. with normal piston speed.

With two cycle engines these bolts are subjected to very slight strain as the connecting rod is chiefly under compression. With the four cycle engines, however, strains arise at the end of the exhaust stroke as the piston is not cushioned and then the pressure due to the momentum of the reciprocating parts may tend to cause the fracture of bolts.

Taking the inertia force at 120 lbs. per sq. in. with a single acting single cylinder four-cycle engine at 150 r. p. m.:

where d = smallest diam. in the bolt thread, then we have

$$2 \times 0.78\, d^2 = \frac{120 \times 0.78\, D^2}{7500} \text{ or } d = \frac{D}{10.}$$

Connecting rods: These are made of various designs. A representative type is shown at Figs. 7 and 8, made preferably of soft steel.

The length of the connecting rod varies slightly in different types of engines. In horizontal engines where a trunk piston is used the length is taken, being not less than five times the radius of the crank, thus the allowable pressure of the piston on the cylinder wall is not exceeded. The length is also governed by the space available for taking out the piston from the cylinder at the crank end. Where a crosshead is used in horizontal or vertical types, the length of the connecting rod may be reduced to 4.5 times the radius of the crank. In engines where the scavenging air pump piston is placed below the motor piston and where the scavenging piston itself acts as

an extra large crosshead, the length of the connecting rod may be 4.25 to 4.5 times the radius of the crank. The wristpin is placed in the scavenging pump piston,

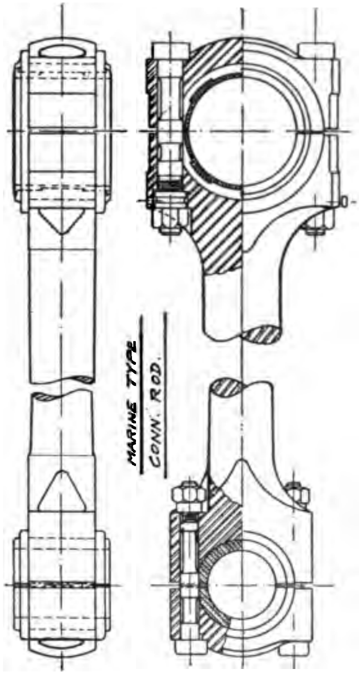


FIG. 7 FIG. 8

and arrangement is made for removing the piston from the back end, first taking off the cylinder head.

The Rod itself should be taken as a long column and its factor of safety is usually taken as 15 to 20. The explosion load $P = 400\, D^2$ tends to crush the rod.

With a safety of 20 the mean diameter will be

$$d = \sqrt{\frac{DL}{40}}$$

where

$L =$ length of con. rod.
$D =$ diam. of piston.

The diameter can be decreased to 0.75 d on the wrist pin end and increased to 1.25 d at the crank pin end.

For a rectangular rod having a height of h and a width of c

$$h = 1.7 \text{ to } 2\,c.$$

$$c = \sqrt{\frac{D \times L}{85.}}$$

AIR COMPRESSORS: An important feature of all Diesel engines is the supply of highly compressed air, necessary for injection into the cylinder with the fuel. With both the 2 cycle and the 4 cycle type engine such air is necessary.

A typical single acting air compressor is shown at Fig. 45. In medium sized engines the 2 stage compressor is sufficient; with larger units a 3 stage compressor is used, the air in any case being cooled between each stage of compression so as to maintain a moderate temperature of the compressed air and thus avoid explosions which might result from the mixture of lubricating oil in air receivers, passages or air connections A double acting type of compressor in the first stage or low pressure air cylinder is also shown at Fig. 9. The compressed air for fuel injection purposes is furnished at 700 to 1200 lbs. pressure.

Some air compressors operate double acting in the first stage where the air is used for scavenging purposes, and single acting in other stages; others

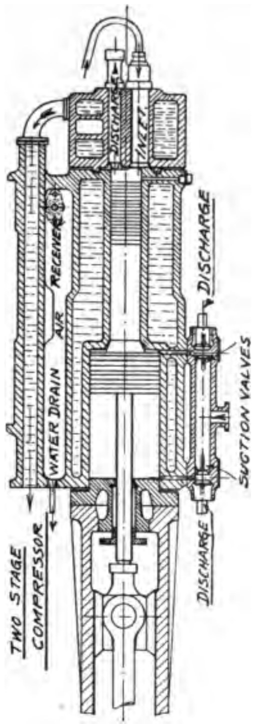


FIG. 9

are all single acting. The compressors are operated preferably directly from the main crankshaft as shown in the various illustrations of different types of en-

gines. They are in some types operated by a rocker arm from the connecting rod as well as by individual motor with the larger units.

Experience has shown that with 100 HP. engine the low pressure stage of compressor has a displacement of 0.3 cu. ft. per minute per HP. This is equivalent to 18 cu. ft. per H.P. hour. Smaller high speed engines require more air than the larger slow speed engines per B.H.P. The power required varies from 8% to 10% of the total power the engine is developing at its full rated capacity. With the two stage compressor the ratio of the volume of the high pressure cylinder to the low pressure cylinder is approximately one to ten. The displacement of the low pressure cylinder of air compressor may be taken as 1/20 of the displacement of the motor cylinder in the four cycle type and as 1/10 in the two cycle Diesel engine.

AIR COMPRESSOR VALVES—The area of the air inlet suction valves for the injection air compressor is usually approximately one-tenth of the area of the piston, and the discharge valve one-seventh that of the piston; in connection with which they each operate this area is free from all obstructions. The compressor cylinders are, of course, properly water cooled.

SCAVENGING AIR PUMPS with the 2 cycle type as explained elsewhere are necessary to expel the products of combustion or scavenge the cylinder by means of a current of air which is usually compressed to

about 4 to 5 lbs. pressure. This is effected in a separate air pump shown in the various illustrations of 2 cycle engines. With some types the pistons in such air pump cylinders are operated directly from the crankshaft, and by other builders the pistons are operated from beam levers or rocker arms actuated from the main connecting rods or crossheads where such are used, the air pumps being placed on the side of the main cylinders. The volume of the air pump cylinder is about 1.4 that of the main cylinder. The ratio of engine stroke to air pump stroke in some instances is 1 to 0.75 or less.

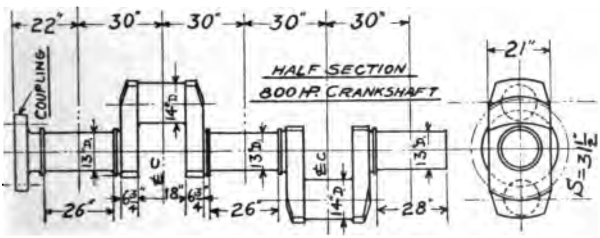


FIG. 10

CRANK-SHAFT:

It is beyond the scope of this work to go fully into the details of the design and construction of such an important part of Diesel Engines as the Crank-shaft. In Fig. 10 is shown a representative crank-shaft for a multi-cylinder engine giving the actual dimensions of each part of the shaft.

For standard Diesel Engines the following formula giving the approximate dimensions of the different

parts of crank-shaft as shown in Fig. 11, and which is partly derived from Güldner, may be used. The ultimate tensile strength of the material is taken as 80,000 lbs. per square inch minimum, the elastic limit being not less than 48,000 lbs. per square inch, with a limit of elongation in 2 in. of 25% and reduction in area being not less than 45%.

$$a = 2.2\,D.$$
$$d^1 = 0.55\,D.$$
$$d = 0.60\,D.$$
$$l = \frac{D}{1.65}$$
$$L = 0.85\,D.$$
$$h = 0.6\,d.$$
$$b = 1\tfrac{1}{4}\,d \text{ to } 1\ 1/3\,d.$$

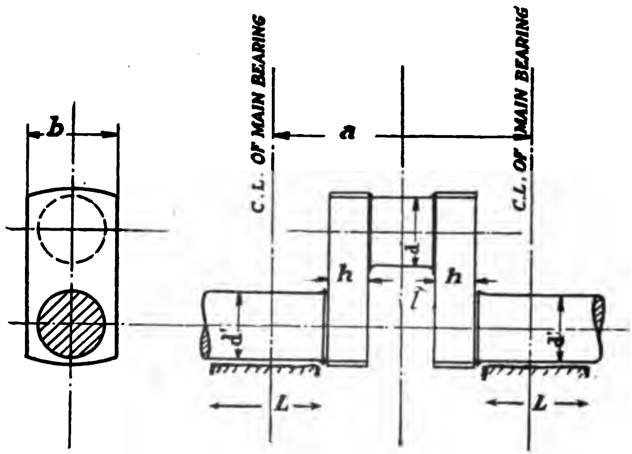


FIG. 11

The dimensions "h" and "b" assume that the crank-shaft has also an outboard bearing.

Mr. James Richardson* in his paper before the Junior Institution of Engineers, Apr. 20, 1914, points out that the following crank-shafts rule of the British Corporation for the Survey and Registry of Shipping, works out very well for most cases for a first approximation:

For crank-shafts, $d^1 \quad \sqrt[3]{\frac{D^2 \times P\,(S + L)}{C}}$

when:

$d^1 =$ diameter of shaft in inches.
$D =$ diameter of cylinder in inches.
$S =$ stroke of piston in inches.
$L =$ length between edges of bearings in inches.
$P =$ maximum initial pressure, lb. per sq. in.
$C =$ constant.

	For Four-Cyle Engines Single-Acting No. of Cylinders.	For Two-Cycle Engines Single-Acting No. of Cylinders.
9000	1, 2, 3, 4	1, 2, 3
8500	6	4
8000	8	..
7500	12	6
7000	..	8
6000	..	6*
4500	..	8*

*"Engineering," page 573, 1914.

*These apply to cases where the cranks are so arranged that two impulses occur simultaneously.

Valve Motion.—In the four cycle type and in the two cycle type where scavenging valves in the cylinder head are employed the valves are operated by means of a vertical shaft actuated by skew gearing from the crankshaft which vertical shaft is again geared to a horizontal shaft, running parallel with the crankshaft and in most engines supported by bearings on brackets attached to the upper part of the cylinders, while in others this shaft is placed lower down. To this shaft are keyed or otherwise attached the various cams required to operate each valve. The motion of the cams is transmitted to the valves through reach rods and levers as shown in the various illustrations. In the Burmeister & Wain marine engine (see Figs. 85 to 89) a chain of spur gears is employed to transmit the power from the crankshaft to the camshaft, and in the Werkspoor marine engine (see Fig. 43) spur gears and connecting rods are used.

Valves:

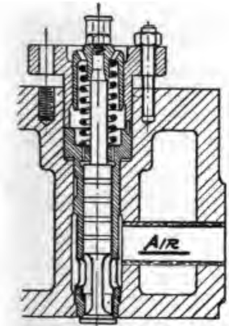
A representative *scavenging valve* is shown in Fig. 12.

The lift of air inlet or exhaust valve $(h) = \frac{d}{4}$

Where $d =$ diam. of valve.

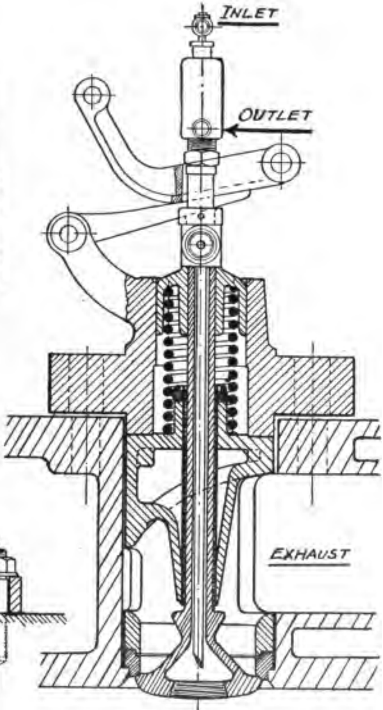
Regarding dimensions of valves and stem. The valve stem $= 0.125\,d + 0.4$. Thickness (t) of valve head.

$$t = 0.15\,d \text{ for steel.}$$
$$= 0.2\,d \text{ for cast iron.}$$



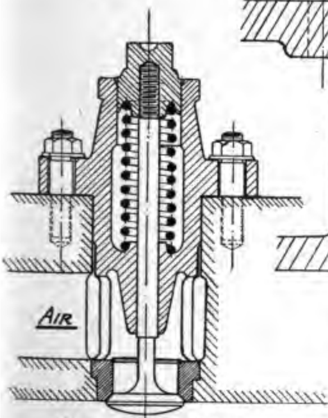
TYPICAL STARTING-VALVE.

FIG. 14



WATER-COOLED EXHAUST-VALVE.

FIG. 13



SCAVENGING-VALVE.

FIG. 12

AN EXHAUST VALVE WATER COOLED as used for engines of 150 H.P. (in one cylinder) or over, is shown at Fig. 13 in section which also illustrates the valve housing separate exhaust seat, water inlet and outlet.

STARTING VALVE. The starting valve through which the compressed air for setting the engine in motion enters the cylinder is shown at Fig. 14. Its dimensions are as follows:

$$f = \frac{F \times c}{70{,}000}$$

Where

f = area of the starting valve in sq. in.
F = area of the piston in sq. in.
c = piston-speed in feet per min.

THE FUEL SUPPLY PUMP as used on many Diesel Engines is shown at Fig. 15. This is actuated by cam or by eccentric from the camshaft with the four-cycle type as shown in Fig. 47 and with the two-cycle type as shown in Fig. 77. A separate fuel pump is furnished for each cylinder in the multi-cylinder engine.

The piston displacement of the fuel pump is such that the amount of fuel pumped is greater than is necessary. Frequently it has a displacement equivalent to four times the amount required, taking the fuel consumption as ½ lb. per B.H.P. per hr. In this way possible leakage of the valves is allowed for, or other improper working conditions, also sufficient fuel is furnished should the engine operate temporarily overloaded.

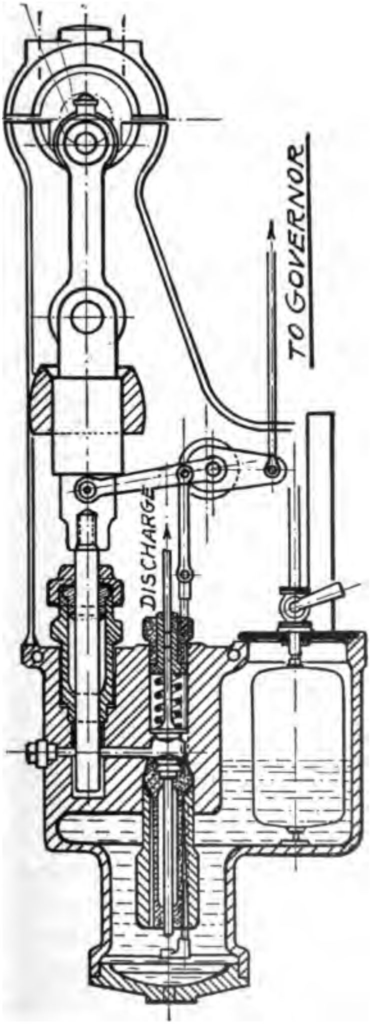


FIG. 15

The pump piston is made of machinery steel hardened, or chilled cast iron. Duplicate valves are furnished on both suction and discharge and are made of hardened steel or phospher bronze. The area of these

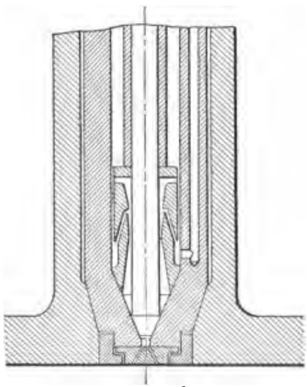


FIG. 16

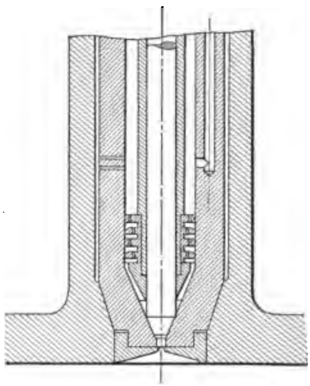


FIG. 17

valves is ample to allow the required amount of fuel to flow past them. The method of governing the speed of the engine where the pump as shown is utilized is simply by causing the governor to hold from its seat for a varying portion of the pump stroke

the suction valve. Thus a varying amount of fuel is delivered to the combustion space in accordance with the load on the engine.

SPRAYERS OR PULVERIZERS.—One of the most important parts of all oil engines is the sprayer or pulverizer through which the fuel is injected into the cylinder

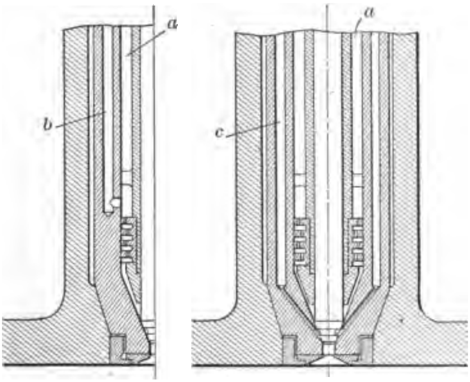


FIG. 18

or compression chamber. A great deal of attention has been devoted in recent years to this part. Sprayers for Diesel engines are shown at Figs. 16 to 22.

Those of more recent design and used with Diesel engines are shown in Figs. 16 to 18. Fig. 16 shows the standard plate sprayer used by many makers and is suitable for lighter oils. Fig. 17 shows this sprayer as made in Sweden. Fig. 18 shows the sprayer adopted by Gasmotorenfabrik Deutz, Augs-

burg-Nurnberg and others where it is necessary to use a slight amount (about five per cent.) of low flashpoint fuel so as to start the ignition more rapidly and allow combustion of the heavy

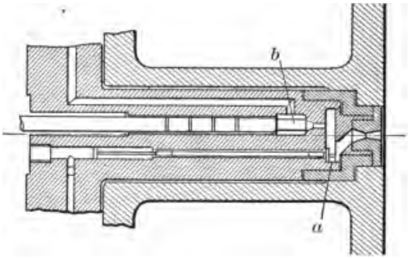


FIG. 19

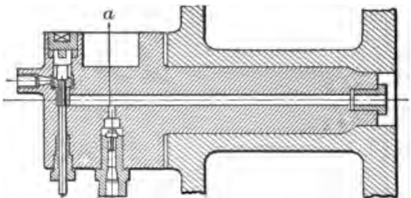


FIG. 20

crude oil or tar oil which is 95 per cent. of the charge to ignite more readily. The method of operation of this sprayer is first, admitting into the cylinder a small quantity of the lighter oil which is followed by the larger quantity of the heavy fuel, two oil injection pumps being used for this arrangement. In this sprayer, Fig. 18, the lighter oil enters through the passage *c*, the heavier oil of higher flashpoint through

the passage *b*, and the lighter oil first enters and passes to the front of the valve. When the valve is raised to allow the heavier oil or tar and air (through *a*) to enter the combustion space, the lighter oil is carried before it and enters first. Being of a lower flashpoint the ignition raises the temperature of compression sufficiently to ignite instantaneously the mixture including the heavier fuel. Fig. 20 shows a sprayer designed for attaching horizontally of the "open nozzle" type, where the fuel enters at *a* and is in direct communication with the cylinder, but further distant than in that shown at Fig. 19. The open type sprayer is shown at Fig. 19, which was invented by Letzenmeyer and as made by Messrs. Koerting. In this arrangement fuel enters the chamber *a*, which is in direct communication with the cylinder, either during the suction stroke or before the compression has advanced. The fuel is injected by compressed air through the opening around the valve *b*, which is opened by the action of the cam controlling it.

SABATHE FUEL INLET VALVE—The fuel inlet valve used by the Societe des Moteurs Sabathe in connection with their engine designed for submarine use is shown in section at Fig. 21.

It is equipped with two valves, one of which is the ordinary type of fuel inlet valve and the second valve is larger in diameter and is loose on the former, being held in place by the spring as shown in the illustration. This fuel inlet arrangement is designed to effect

"mixed combustion" by allowing two periods of fuel injection. Its operation is as follows:

When the engine is operating at lighter loads fuel only enters by the lower passage through the smaller

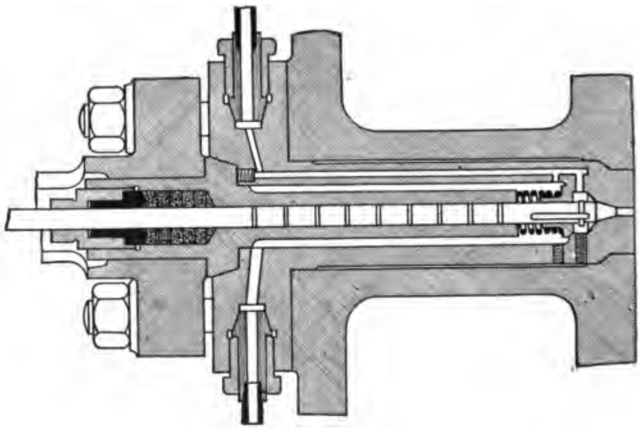


FIG. 21

valve and no secondary injection from the other valve and passages is allowed. The fuel injection from the first valve is arranged to enter the combustion space when the compression is about 450 lbs. and the volume is constant, with the result that the pressure then instantly rises. Immediately afterwards the injection of fuel from the upper passage hitherto held in check by the larger valve is allowed to slowly enter the combustion space while the volume is increasing. The movement of the valves is, of course, mechanically

controlled and the timing of injection can be altered to suit the requirements of varying speeds.

Economy of injection-air as well as greater efficiency or lower fuel consumption is claimed for this type as itt chief advantages.

THE M. A. N. HORIZONTAL FUEL INJECTOR is shown at Fig. 22. This design is suitable for the use of tar oils and when a very small amount of ignition or lighter oil is also used. In operation a slight amount of the lighter fuel is fed first to the chamber closest to the valve seat; the heavier oil passes only to the main injection chamber. In this sprayer baffle cones have replaced the usual form of atomizing plates. The air passages on the upper side of the valve spindle are less in area than those on the underside, thus a greater velocity of air is found in the upper passages than in the lower passages, when the valve is opened and air at high pressure passes through them. The ignition or lighter fuel being in closest proximity to the valve, when it is opened this lighter fuel naturally first passes through the sprayer and is ignited thus; the heavier oil or tar when it afterwards enters the combustion space it comes in contact with gases at a higher temperature and consequently its ignition is more easily and surely attained.

LUBRICATION.—In Diesel engines of all types for marine work, particular attention has been paid to the arrangement of the lubrication. For the piston, special

lubricating oil having a high flashpoint and with a very small percentage of animal oil is used. It is furnished

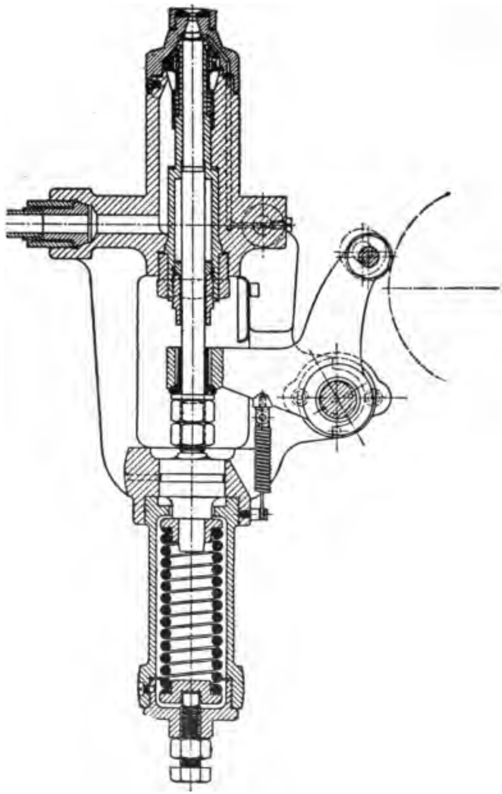


Fig. 22

by a positively operated force feed oil pump actuated from the camshaft or other moving part of the engine, preferably by separate pumps for each piston. The

oil is delivered through separate copper pipes to different parts of the piston and cylinder surface, thus ensuring proper distribution of the lubricant.

The main or crankshaft bearings are furnished with a plentiful supply of oil which, in the later designs of engines is delivered by gravity and is forced around and on to the bearings. Then it is conducted through an oil filter and to a special tank in which is a cooling water coil, and after proper cooling descends to the sump to be pumped through the bearings again.

The piston pin is lubricated either by a sliding tube placed on the piston or crosshead which is arranged to deliver the oil directly to the piston pin or in some designs lubricating oil is fed by pressure pump through the crankshaft which is then made hollow. The lubricant is forced on to the surface of the crankpin bearing and is conducted through a hollow space in the connecting rod up to the piston pin.

BRAKE OR ACTUAL H.P. To ascertain this power of an existing Diesel Engine, the following formula can be used:

1. Four-cycle engine, single acting.

$$N = \frac{F \times s \times n}{880}$$

or if the B.H.P. of the engine is known, then the dimensions of the cylinder are found as follows:

$$F \times s = \frac{N \times 880}{n}$$

2. With the 2-cycle single acting engine

$$N = \frac{F \times s \times n}{500}$$

and similarly the cylinder dimensions are found:

$$F \times s = \frac{N \times 500}{n}.$$

Where

$N =$ B.H.P.
$n =$ revolutions per minute.
$F =$ piston area in sq. in.
$s =$ piston stroke in feet.

The ratio of the cylinder diameter to the stroke varies from 1:1 to 1:2.

A good ratio for stationary engines is 1 : 1.5.

CAMSHAFT. This is made of cold rolled steel having

$$\text{Diameter} = \frac{D}{5.5}$$

where

$D =$ diameter of cylinder.

AIR TANK FOR INJECTION. Where separate tank is used to contain the air for injecting with the fuel, its capacity should be equivalent to approximately 0.035 cu. ft. per B.H.P.

AIR TANK FOR STARTING may have capacity = 0.35 cu. ft. per B.H.P.

FLYWHEELS. The flywheel of a Diesel Engine made in two halves is illustrated at Fig. 23. In this design

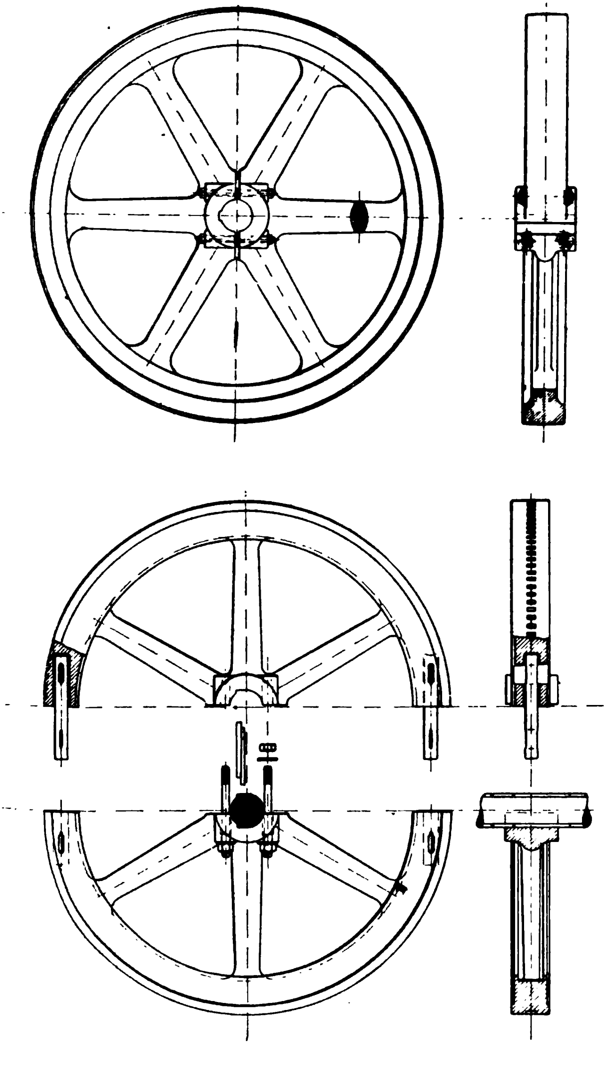


Fig. 23

the two halves are connected by a circular link connected to the rim, as shown in illustration. Another method of connecting the two halves of the wheel is shown at Fig. 24.

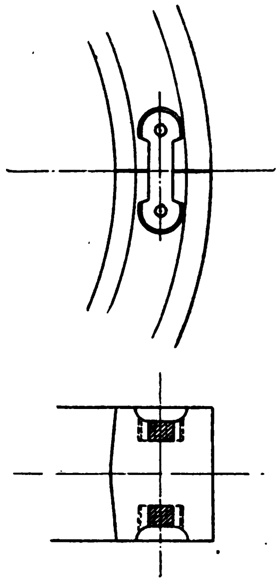


FIG. 24

The flywheel with a stationary engine is of different weights, varying with the different types of engines and the degree of unsteadiness from the uniform speed of rotation allowable in different installations.

The function of the flywheel is to maintain an even speed of rotation during each cycle of operation of the engine. Therefore its weight will vary in-

versely with the number of impulses given per revolution of the crankshaft.

While the flywheel with a four-cycle single-acting single-cylinder Diesel Engine will necessarily be heaviest, as the number of cylinders are increased or the impulses per revolution are increased the weight of the flywheel can be decreased to give the same effect. The governor controls the speed of the engine and the flywheel controls the cyclic variation and the degree of unsteadiness from uniform speed of rotation during each cycle.

Let T = Degree of unsteadiness.
Then

$$T = \frac{V\,max. - V\,min.}{V\,ave.}$$

Where
$V\ max$ = max. velocity of shaft during cycle.
$V\ min$ = min. velocity of shaft during cycle.
$V\ ave$ = Average velocity of shaft during cycle.

The value of T may be taken as follows:

For factory use....................	1 : 30 to 1 : 40
For d.c. electric generators direct connected to engine crankshaft.....	1 : 100 to 1 : 120
For a.c. generators direct connected..	1 : 175 to 1 : 200

THE PERIPHERAL SPEED of the flywheel should not exceed 6,000 feet per min. for cast iron. In computing its weight the hub and arms of the wheel are neglected in the following formula:

$$W = C \frac{N.}{D^2.\ T.\ n^2.}$$

Where

$W =$ weight of Rim in tons (2000 lbs.).
$D =$ dia. of rim (in ft.) at center of gravity.
$n =$ Revs. per min.
$N =$ actual or B.H.P.
$C =$ constant.

Güldner gives the following values of C:

$C = 712{,}000$ for 4-cycle single-acting with impulse each 720°.

$C = 312{,}000$ where the impulse is each 360°.
$C = 178{,}000$ with impulse each 240°.
$C = 45{,}000$ with impulse each 180°.
$C = \ \ 6{,}080$ with impulse each 120°.

HUB OF FLYWHEEL. The dimensions of the hub as shown in illustration may be taken as follows:

Length = 1.5 to 2.5 of diameter of shaft.

The outside diameter of the hub may be taken as 2 to 2.5 diameter of shaft.

ARMS OF FLYWHEEL. It is difficult to give a formula for the arms of the flywheel suitable for all cases. The equivalent of "W" should be figured for different engines. Professor Torrey gives the following formula:

$$W = \frac{S v}{n}$$

$$b = \frac{W L}{30 d^2}$$

Where

W = load in pounds acting on one arm.

S = strain on belt in pounds per inch of width, taken at 56 for a single and 112 for double leather belts.

V = width of belt in inches.

v = number of arms.

L = length of arm in feet.

b = breadth of arm at hub in inches.

d = depth of arm at hub in inches.

AIR INLET PIPE SIZES.

$$f = \frac{F \times c}{60}$$

Where

f = area of the inlet pipe in sq. in.

F = area of the piston in sq. in.

c = piston speed in feet per second.

EXHAUST PIPE SIZES. Between the engine and the silencer this pipe should have an area of 1.15 to 1.3 times the area of the exhaust valve. From the silencer to the atmosphere the area of the exhaust pipe can be slightly decreased. Arrangement should always allow for the expansion and contraction of the exhaust pipe, which is frequently as much increased as 1/16 to the foot when fully heated.

COOLING.—A sufficient supply of cooling water to maintain the proper temperature of the cylinder is necessary to circulate around its water jacket—three

to four gallons of water per B. H. P. hour which should not exceed an outlet temperature of 175° F. with the smaller diameter four cycle type. With the large diameter cylinders and of the two cycle type five to ten gallons of water per B. H. P. hour is required and the outlet temperature should not exceed 120° F. In the larger four cycle engines the exhaust valves are also water cooled, being made hollow, the cooling water entering and leaving through the hollow valve stem or guide. In the two cycle type and larger sizes of the four cycle type engines the piston is provided with a space for cooling water or cooling oil at the combustion end, which liquid is conducted to and fro through sliding telescopic tubes or swivel jointed tubes. Provision for cooling the main crankshaft bearings is also made either by direct water cooling or by a system of cooling the lubricating oil referred to later.

COOLING WATER INLET PIPE.

$$d = \sqrt{.02 \times N}$$

Where

$d =$ diameter of the pipe.
$N =$ B.H.P. in each cylinder.

The outlet water pipe should be 1.5 of the diameter of the inlet water pipe.

PISTON SPEED. Examination of the various Diesel engines now built shows an average piston speed of 800 ft. per minute. A piston speed of 900 to 1,000 ft.

per minute is used in some engines. Nevertheless, 800 ft. per minute is considered good practice. Thus the film of lubricating oil at this speed is thoroughly preserved and with it minimum leakage of piston or piston rings is found. The chief advantage of high piston speed is of course the reduction of weight in the engine and consequently a reduction in cost of manufacture. With the four-cycle type, increased speed entails provision of greater valve areas so as to avoid throttling of air inlet or exhaust. With the two-cycle type increase of piston speed requires provision of greater pressure of scavenging air so as to insure sufficient supply passing the scavenging valves, but increasing the pressure of scavenging air again entails greater loss of power.

Bed plate and frame. The material for these parts should be good quality of cast iron. The thickness of metal should be such that the maximum strain is not over 3,000 lbs. per sq. in. The pressure on the foundation on which the engine stands should not exceed 50 lbs. per sq. in., taking total weight of engine.

Foundation should be built of good concrete, mixture recommended for which is one part Portland cement and six to seven parts sand and broken stone.

For horizontal engines the weight of foundation should not be less than 2,000 lbs. per B. H. P. with single cylinder engine and slightly less for twin cylinder engine.

With vertical multi-cylinder engines this weight can

be slightly decreased. Güldner gives with the two-cylinder engine 1,750 lbs., with three-cylinder 1,300, and with a four-cylinder engine 1,100 lbs. of concrete per B.H.P. of engine. The depth of the foundation should not be less than 5 D where D = diameter of cylinder.

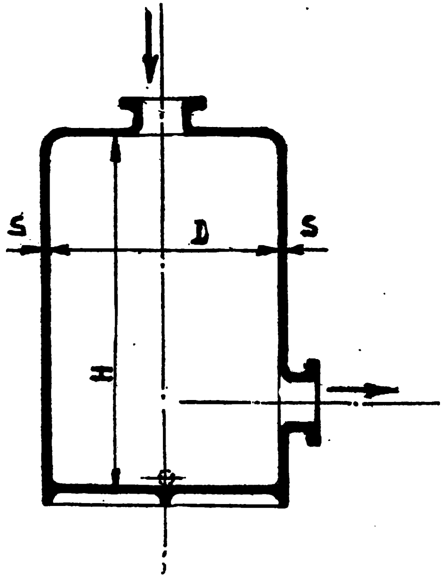


FIG. 25

The above figures assume that the foundation is built on solid ground. Where quicksand or wet ground is encountered, piling of course is necessary, or proper preparation of such ground must be made to make it equivalent to solid ground before the building of the foundation is commenced. It is usual to thoroughly

insulate the foundation of the engine from foundations of the building in which the engine is placed and which are in close proximity to the engine foundation.

SILENCER. This is made of cast iron as shown in Fig. 25. Its dimensions are such that its cubical area is equivalent to at least six or eight times that of the piston displacement.

$$H = 1.25 \text{ to } 1.75D.$$

$$S = \frac{D}{50} + \frac{1}{4} \text{ inch.}$$

$$S \geqq \frac{1}{2} \text{ inch.}$$

Where complete silencing of the exhaust is required, the silencer is made larger, sometimes having a cubical area equivalent to fifteen times the displacement of the piston, or in that case a concrete silencing pit is used.

Figs. 26 to 29 left for future additions.

*Some of the formulæ have been suggested by Mr. Fritz Gruetzner, V. d. I., and practically all have been tested and corrected by him.

CHAPTER III

INDICATOR CARDS.*

The following diagrams are taken from various two and four cycle Diesel engines. The particulars available of each diagram are stated in the following paragraphs. Each diagram is numbered and the paragraph giving information concerning it is similarly numbered. The scale nearest to the card is—kilogramm per square centimeter. The scale further from the card is—lbs. per square inch.

No. I.

Diagram from 4 cycle single acting vertical multi-cylinder marine type engine: Fuel—Borneo crude, M. E. P.=92 lbs. per sq. in., R. P. M.= 100, Cylinder diameter= 29 9/64 in., Stroke=43 5/16 in.

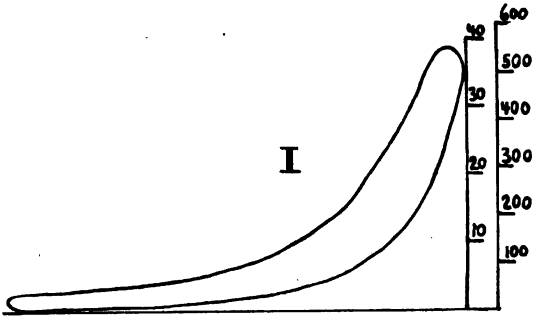
No. II.

M. E. P.=90 lbs. per sq. in., R. P. M.=252—full load from 4 cycle vertical marine type engine.

No. III.

M. E. P.=62 lbs. per sq. in.—half load. Otherwise same as No. 2.

*Full instructions for the use of the indicator, measurement of indicator diagrams as well as detailed instructions for testing oil engines are given in the author's treatise "The Design and Construction of Oil Engines."


I
40
30
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600
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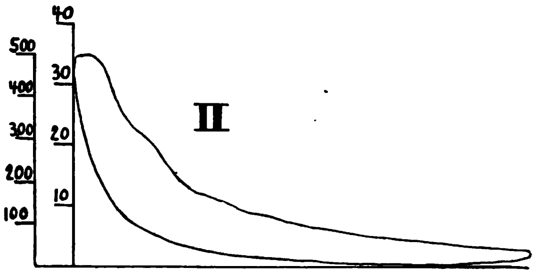
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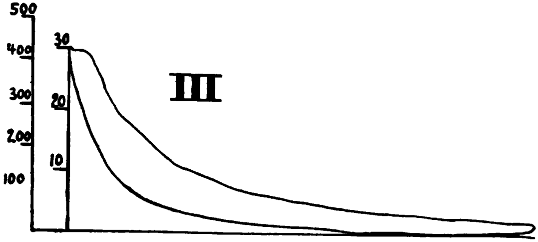
III
500
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10

No. IV.

Two cycle horizontal single acting: M. E. P.=75 lbs. per sq. in., injection air pressure=800 lbs. per sq. in., scavenging air pressure=4.7 lbs. per sq. in., R. P. M.=120, cylinder diameter=480 mm. (18 13/16 in.), stroke=710 mm. (27 15/16 in.). Approximately two-thirds load. The diagram shown by dotted line is taken by manipulating the indicator by hand. It shows in an extended form the compression and combustion pressure. The point at "a" indicates the compression while the line at "b" indicates the ignition pressure. The expansion continues as shown by the dotted line towards the right.

No. V.

Taken from the same engine as No. 4, the indicator being fitted with a stop which is shown by the dotted line on the upper part of the indicator.

No. VI.

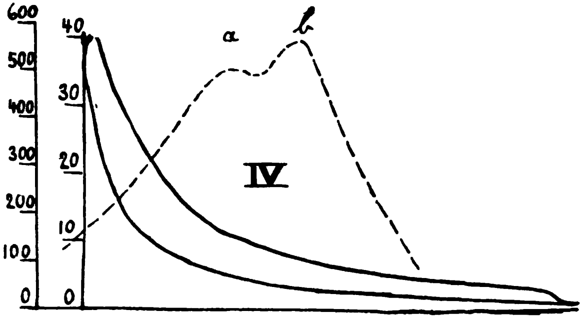
Taken from same engine as No. 4 with a very light spring, the indicator being fitted with a stop. It shows in the 2 cycle type the exhaust and air inlet.

No. VII.

Taken from the same engine as No. 4 with a still lighter spring than No. 6 and shows more plainly the conditions existing during part of exhaust and air inlet.

No. VIII.

Taken from same engine as No. 4 to No. 7. Approximately full load. The spray valve opens too early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200
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20
10
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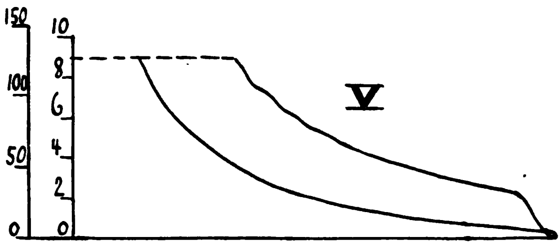
a
b
IV


150
100
50
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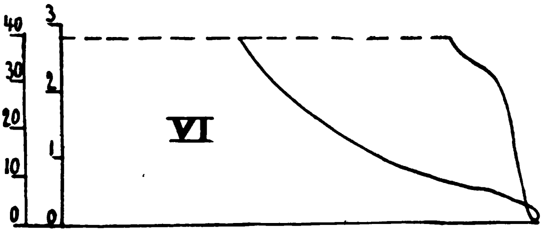
10
8
6
4
2
0
V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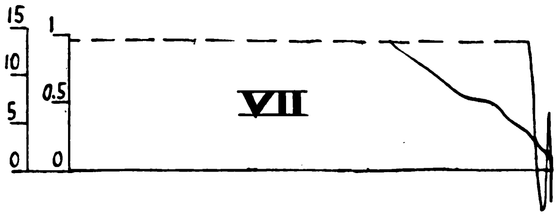
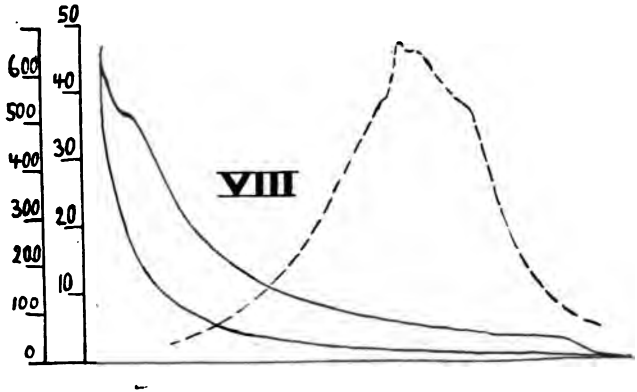
30
20
10
0
3
2
1
0
VI


15
10
5
0
1
0.5
0
VII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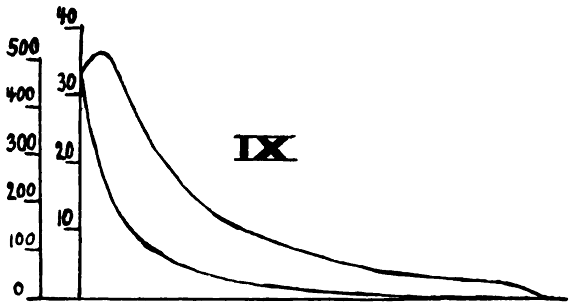
VIII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IX

and the defective operation of the engine is shown by the incorrect loop at the ignition. This requires later period injection. The dotted line shows more plainly the operation pressure similar to No. 4.

No. IX.

Taken from "Junkers" type engine. R. P. M.=120, pressure of injection air=1,000 lbs. per sq. in., pressure of scavenging air=2.1 lbs. per sq. in. Taken about full load.

The low pressure of scavenging air required with this type of engine is remarkable. The combustion line is ordinarily straight. In this diagram too early injection of the fuel is indicated.

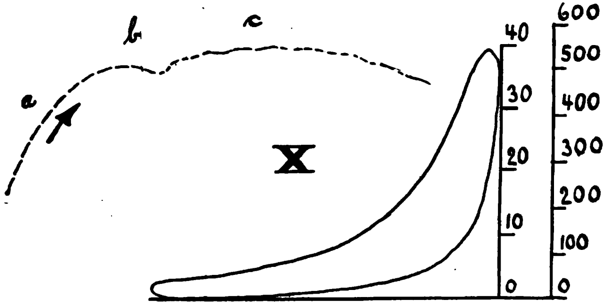
No. X.

Taken with scale 1″=371 lbs.

Four cycle vertical type 18⅞ diam., 28⅜″ stroke—full load. The dotted line is taken by hand similar to that shown on No. 4; "a" indicates the compression pressure, "b" shows the pressure with the piston at the upper centre, "c" indicates the ignition and expansion.

No. XI. and XII.

Diagrams taken from the same engines as No. 10, but with indicator spring being 1″=360 lbs. and with varying loads.


a
b
c
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500
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200
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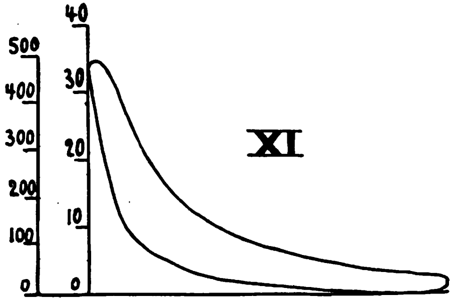
0

X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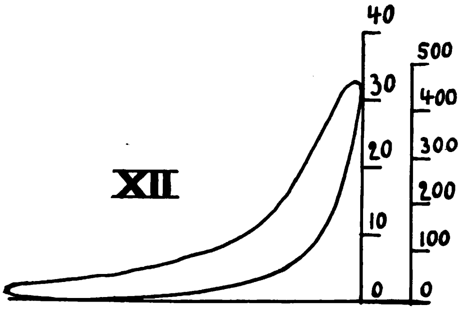
XI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100
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XII

CHAPTER IV

MARINE DIESEL ENGINES DISCUSSED

There are nearly 500 ships now operating propelled by Diesel engines, and numerous engineering firms well known for the superiority of their output in nearly all countries of Europe are engaged in building them, while in this country not so many firms have yet undertaken their manufacture.

ADVANTAGES AND DISADVANTAGES.—Some of the advantages of this engine for marine purposes are:

1. The space occupied by it is less than that required for the steam engine and boilers with consequent greater space in the ship available for cargo.

2. The amount of attention required is less. The stokers and coal trimmers necessary with the steam engine ships being reduced in number if not entirely eliminated.

3. The facility for storing the fuel for the oil engine as compared with that of the coal necessary for a steam engine.

4. The greater distance that a ship equipped with the oil engine can travel as compared with the steam engine because less fuel is used by it.

5. The absence of large funnels of the steam engine and the elimination of smoke.

6. The quick starting of the engine which can be accomplished at a moment's notice.

7. Elimination of standby losses—that is, as soon as the engines are stopped the fuel consumption ceases.

8. Replenishing the store of fuel. At sea the coaling of a steamship is impracticable whereas oil fuel can, if necessary, be transferred at sea.

The amount of coal, varying with its quality, consumed in a steamship for propelling purposes only, is somewhat over 1½ lbs. per I. H. P. per hour or 1.8 lbs. of coal per B. H. P. per hour. These figures represent the best conditions and probably a fuel consumption of 2 lbs. per B. H. P. per hour would be a fair estimate. The amount of liquid fuel used in a Diesel engine may be taken as 0.4 lbs. per B. H. P. hour. Consequently, the weight of fuel consumed in the Diesel engine as compared with the steam engine is about one-quarter to one-fifth. Again, when the engine is running at a reduced speed the relative economy of the oil engine would then be greater than with the steam engine.

The coal must be placed in a position accessible to the boilers; liquid fuel can be placed so that the space occupied by it does not interfere with the storage of the cargo. This again increases the earning capacity of the vessel. It is estimated that on a cargo steamship equipped with reciprocating engines and boilers the weight is about 300 lbs. per I. H. P., possibly 250 lbs. for turbine propelled boats. The weight of a Diesel engine including all accessories would be approximately 150 lbs. per I. H. P. and high speed engines both of the steam or the Diesel type would each be respectively nearly one-half of the weights above given. The space occupied by the Diesel engine is

about the same as that occupied by the steam engine alone—thus the space occupied by the boilers is free in the Diesel engined ship and is available for cargo or other purposes.

DISADVANTAGES.—Some of the disadvantages of the Diesel engines for ships may be stated as follows:

1 Reliability—the marine steam engine has been in operation for generations—most engineers are thoroughly conversant with it. The Diesel engine is comparatively new and unknown by marine engineers. It must have special care and attention. With improper handling and even with some derangement the steamship can be temporarily repaired and brought into port, whereas the Diesel engined ship under the same conditions and with the same handling might be helpless.

2. The Diesel engined ship unquestionably requires a high grade of attention, more so than does the steam engined ship, which class of help may not be available and difficult to replace (in case of sickness or casualty) in foreign ports.

3. Owing to long experience of present marine engineers the steam engine can be adjusted and kept in proper operating condition more easily than can the Diesel engine.

4. Troubles with the steam plant can be more easily investigated and remedied than with an internal combustion engine, especially if it is in the hands of an inexperienced or careless or untrained attendant.

5. Maintenance of a plentiful supply of compressed air for starting and manœuvering.

Many of these disadvantages will disappear or become unimportant as the Diesel engine becomes better known to marine engineers but they are worthy of consideration at the present time.

TYPES.—The Diesel marine engines have been built as follows:

1—Four cycle single acting.
2—Two cycle single acting.
3—Two cycle double acting, and
4—Junkers engine.

For land purposes the four cycle engines have been built in the vertical type for slow and high speed and also in horizontal single acting and double acting type. The two cycle engine is also built for slow as well as high speed vertically and horizontally single acting.

For marine purposes, of course, only the vertical types are built, and they are made non-reversible and directly reversible. The four cycle engine has hitherto been chiefly used for land purposes. Greater experience has been gained with it for marine purposes also, and it has been thus used with satisfaction in smaller sizes. The tendency toward building the Diesel engine in larger sizes has brought about the desirability of the two cycle type. It has been found impracticable to build the four cycle cylinder of the large dimensions that would be required, and accordingly the only method of increasing the capacity of the engine was to multiply the number of cylinders. With the four cycle

type this has proved complicated on account of the increased number of moving parts and more numerous valve motions, etc.

For engines of over 1000 H. P. the two cycle type has found greater favor. Cylinders of over 1000 H. P. have been constructed and plans have been made for such of even larger sizes. The two cycle single-acting type, on account of its comparative simplicity and the absence of piston rods and stuffing boxes has hitherto been preferred to the two cycle double-acting type.

The two cycle is capable of developing nearly double the power of the four cycle with cylinders of the same dimensions, at least, the power in the two cycle engine is increased about 75 per cent. over that of the four cycle. On the other hand, the four cycle type is slightly more economical than the two cycle, the fuel consumption being 0.4 lbs. in the four cycle and .45 lb. in the two cycle. In the four cycle type usually more complete combustion of the fuel is obtained and a somewhat lower grade of fuel can be utilized. For the larger size engines, that is those over 1000 H. P., the two cycle type has some advantages over the four cycle in that it requires less space, its weight should be also less.

CHAPTER V

OPERATION AND CORRECTION

PRELIMINARY—The engine room properly arranged allows sufficient space around the engine for its proper attendance and for the removal of parts for inspection or repair. If of the vertical type, sufficient head room is provided for withdrawing the piston and connecting rod or other parts which occasionally require removal. An overhead crane should be installed, which is advantageous not only during installation, but when occasionally repairs are necessary or the engine has to be taken apart. The engine room should be well lighted and should be dry, free from injurious gases and dust. The floor preferably should be of dustproof material. Care should be taken that the engine room be properly heated, especially in cold climates, so that freezing of the water in the circulating water pipes or passages is not possible.

OPERATION—The durability and cost of maintenance of a Diesel engine depend very largely on careful attendance and the remedying of any defect directly it is shown. A Diesel engine to give its maximum efficiency should be kept in the best possible working condition.

LUBRICATION—Three qualities of lubricating oil are frequently used, one for the main cylinder and another

of very high flash point for the air compressor cylinder and pistons, the other for all bearings. With some types grease is also necessary for the lubrication of some of the parts. All the parts of the engine requiring lubrication must be kept perfectly clean. The main piston is lubricated with special oil of good lubricating quality, chiefly mineral, free from acids or animal matter having flashpoint of not less than 500° F. While sufficient lubrication must be given to both main piston and the compressor pistons, more than sufficient lubrication is detrimental; with the main piston it causes possible sticking of the rings, and with the compressor pistons it may gasify, and mixing with the compressed air form an explosive mixture and might cause an explosion in air pipes or tanks; it also entails clogging of compressed air passages, fuel inlet valve and sprayer. Care should therefore be taken to reduce the lubrication of compressors to the lowest limit possible. Gearing, rollers and cams can be advantageously lubricated with grease.

IF AN OIL FILTER is used the filtered oil should be employed only on the bearings. New oil should be used on air compressor and piston. Care should be taken that all lubricators are in proper operating condition before the engine is started and after the engine is stopped, all lubricators should be closed to prevent waste of oil.

STARTING THE ENGINE—It is well if any repairs have been made since the previous working of the en-

gine, to turn it over once or twice by hand to be assured that the moving parts are clear and that no obstruction can prevent its proper operation. The following instructions must be treated as general only. Detailed instructions applicable to each particular engine are furnished by the makers and should be carefully studied in addition to these remarks. In a general way the operation of starting the engine can be effected in accordance with the following rules for stationary engines:

a. *Fill all lubricating devices* with the proper lubricants already referred to and care should be taken that they are each in action so that lubrication commences before or immediately the engine starts to operate.

b. *The cooling water supply* should be properly furnished to all parts of the engine requiring to be cooled. If necessary, prime the circulating water pump where such is used and open all the valves for the proper outlet of the cooling water.

c. *The engine* should be placed, if necessary, in proper starting position by barring over the flywheel or movement of same by other means if provided.

d. *The exhaust cam* should be placed on the low compression side or otherwise arranged as required on engines when such provision is made.

e. *Air valves.* Care should be taken that the injection air valves are in proper operating condition.

f. *Manipulate* the handle controlling the starting valve so that it is in the proper position for starting.

g. *Note the pressure* in starting air receivers that it is sufficient.

h. *Open wide* the air inlet to the low pressure side of air compressor.

STARTING PROPER—Where automatic starting arrangement is furnished, open the valve on the starting air receiver and the valve on the injection air receiver (the fuel supply to the sprayer having previously been furnished by hand).

a. *When the engine* has attained sufficient speed, the starting valve lever is put out of action and the receiver containing the starting air shut off, the injection air valve being opened wide.

b. *The fuel supply* can be fully opened. Where a light and heavy fuel are used the heavy fuel can now be brought into action. If, however, this heavy oil requires heating, care should be taken that the cooling water issuing from the cylinder has attained sufficient temperature to heat it to about 110° F.

c. *Low pressure starting.* If at any time it is necessary to start the engine with comparatively low pressure air (200 to 300 lbs.) this can be accomplished, but care should be then taken before starting to see that all parts are in proper working condition so as to avoid loss of air which would necessitate a second start. Engines can be started with pressure in the air receivers as low as 250 lbs. If, however, the injection air pressure is below 600 lbs. the valve on the injection air receiver should be closed when the starting air receiver valve is opened. It can be reopened as soon as the engine has attained its full speed. Sometimes compressed carbonic acid gas is used for starting where

compressed air is not available, but *compressed oxygen is dangerous and should never be used.*

d. *Low temperature.* If the engine is started where low air temperatures exist it may be necessary to fill the cylinder water jacket with heated water or heat them with steam and also reduce the friction of the engine by the use of heated oil, on the bearings, etc.

REMARKS ON OPERATION

a. *Charging Air Receivers.* Bring the pressure in the starting air receivers up to normal as quickly as possible.

b. *Injection Air.* The pressure of the injection air is important. Too low injection pressure may be the cause of poor combustion or late ignition in the cylinder. Too high injection pressure is indicated by improperly timed or early ignition and possibly slight knocking.

c. *Cooling Water.* Care should be taken that the cooling water circulates through all parts requiring it. If thermometers are furnished note carefully the temperature. One hundred and sixty degrees F. should be the maximum with a 4 cycle engine and with a 2 cycle engine 120° F.

d. *Bad Water.* The water for cooling purposes should always be analyzed. If it is used continuously and recooled and the analysis shows it necessary, the water should receive treatment. In any case care should be taken that deposits of lime, etc., cannot take place in the cylinder jackets.

Graphite can be advantageously used on all gaskets exposed to heat.

SCAVENGING AIR—It is very necessary to observe that the air pressure in the air reservoir or receiver of the scavenging air with the 2 cycle type is always maintained at approximately 4 lbs. This pressure varies with different makes and is mentioned in the directions. The pressure failing is due to leaky valves or other leakage, and the cause must be ascertained and remedied immediately.

WATER OR OIL COOLED PISTON—Before starting the engine make sure that the supply of cooling medium to the piston is in proper working order. Immediately the engine is started the cooling supply should begin to circulate to the piston. Care must always be taken that this supply continues in proper condition while the engine is operating, otherwise carbonizing of the lubricant will take place, resulting in piston rings sticking and failure of proper operation.

INJECTION VALVE CLEANING—When the engine is stopped it is an excellent practice to force the inlet valve or spray-valve open and inject a small quantity of kerosene around it and in the chambers of the spray valve. This procedure effectually cleans the valve and facilitates starting the engine next time.

IMPROPER OPERATION

FAILURE TO START—If trouble is encountered in starting the engine the cause is frequently due to the

starting valve not working properly and it should be examined to ascertain if it is in proper working order.

Ignition Failing—This may be due to several causes. In the first place, the pressure of compression through leakage past piston rings or air inlet or exhaust valves, and thus the temperature caused by compression may not be sufficient to properly ignite the fuel. Or the fuel oil supply itself may be inadequate. In that case, test each part of the fuel supply separately and make certain that a proper amount of fuel is delivered to the sprayer. Or, again, the timing of the fuel entering the cylinder may be inaccurate. In this case the mechanism controlling the fuel injection should be examined. Sometimes an air pocket may be found in the fuel oil supply pipe preventing the proper supply of fuel to the sprayer. In other cases improper operation has been occasioned by leakage of the pump suction or discharge valves. Again, leakage past the pump plunger may occasion trouble. Still another cause of improper starting may be the lack of fuel in the oil pipe delivering fuel to the sprayer; opening the valve usually placed in the oil piping near the sprayer will show if it contains fuel and is free of air. Fuel must never be pumped up by hand when the engine is running.

It is essential that proper fuel supply is at all times maintained; the fuel tanks and all connections between them and the fuel injection valve should be thoroughly clean and perfectly tight without possibility of air pockets in the fuel delivery pipe.

LEAKAGE past the piston at any time (and especially if at starting flame is seen to issue from around it) is probably due to the piston rings sticking. They should in such case be examined and cleaned at the earliest opportunity by removing the piston from the cylinder and thoroughly cleaning the piston rings with kerosene so that they are free to move in the piston grooves.

THE PRESSURE OF COMPRESSION must at all times be maintained. If leakage is suspected and it cannot readily be detected, the indicator should be applied and a few diagrams taken, and if a lower pressure than the normal is shown to exist and if the piston works without leakage, then each valve requires careful examination and should be reground to its seat if necessary. Leakage around the valve housing or around the valve seat may be the cause of the trouble.

COLOR OF EXHAUST GASES—It is important to note that the exhaust gases are emitted of correct description. If the gases are smoky or black this indicates that improper combustion is taking place due to leakage, as already referred to, or that too much fuel is being used or that the injection air is not correct. If the engine is working correctly the exhaust gases will be nearly or quite invisible.

SMOOTH RUNNING—A Diesel engine if properly adjusted and in correct operating condition will work smoothly, quietly and evenly. Knocking or uneven rotation may be due to either (a) the timing of fuel

injection being incorrect, that is, it reaches the combustion space too soon, and the ignition commences before the proper time, or it is too late. (b) Knocking may be due to loose main bearings, or the connecting rod bearings may require adjustment. Careful examination of the bearings when the engine is stopped by placing the hand partly on each bearing and partly on the journal while slight movement by hand is made, will show the attendant if the bearings, and which bearing, requires adjustment. Noisy operation in the cylinder may be due to improper action of the fuel inlet valve, as wrong timing of its opening may cause slight explosions instead of proper combustion at a slower rate. See also that the valve is allowed to close properly, also that no leakage is taking place.

LIGHT LOAD—When a multi-cylinder engine is running at light loads ignition may cease in one of the cylinders, which may be due to lack of fuel supply. If the load on the engine is so light and remains thus continuously, and it cannot be remedied otherwise, then one or more of the cylinders, if in a multi-cylinder engine, can have the fuel shut off entirely, thus increasing the load on one or more cylinders.

OVERLOAD—Faulty operation is sometimes due to the engine being overloaded. If it is necessary to continue the operation of the engine in this condition (which is frequently indicated by a smoky exhaust and by the engine laboring) and the combustion is indicated to be not as complete as it should be, then sometimes rais-

ing the pressure of injection air will remedy or partly remedy this trouble. It is possible that slightly earlier injection will help. On the other hand, the trouble may be due to leaky fuel inlet valve or leaky air or exhaust valves.

TEMPORARY OVERLOADING may again be due to increased friction caused by hot bearings or overheated piston. Careful investigation will frequently locate trouble of this kind and, of course, increased lubrication temporarily until the trouble is overcome should be allowed.

UNEVEN RUNNING—If the engine runs unevenly and does not maintain its proper speed, this may be due to the governor being out of order or leaky fuel pump valves. The regulation of the governor should be carefully noted and if it is the cause of the trouble, eased so that even operation results. In other cases when the engine is operated improperly the injection air pressure will vary. This is due to the air compressor pistons leaking or the automatic air valves leaking or sticking and not seating properly. Sometimes in a multi-cylinder engine, each cylinder will be found not to be doing its share of the work. This may be caused in an engine, which has individual fuel pumps for each cylinder, by one or more of such pumps not operating efficiently due to the valves in the pump not seating or to leakage. The exhaust from each cylinder should be noted, and if they are all similar in sound and color, this will indicate that each cylinder is doing its share.

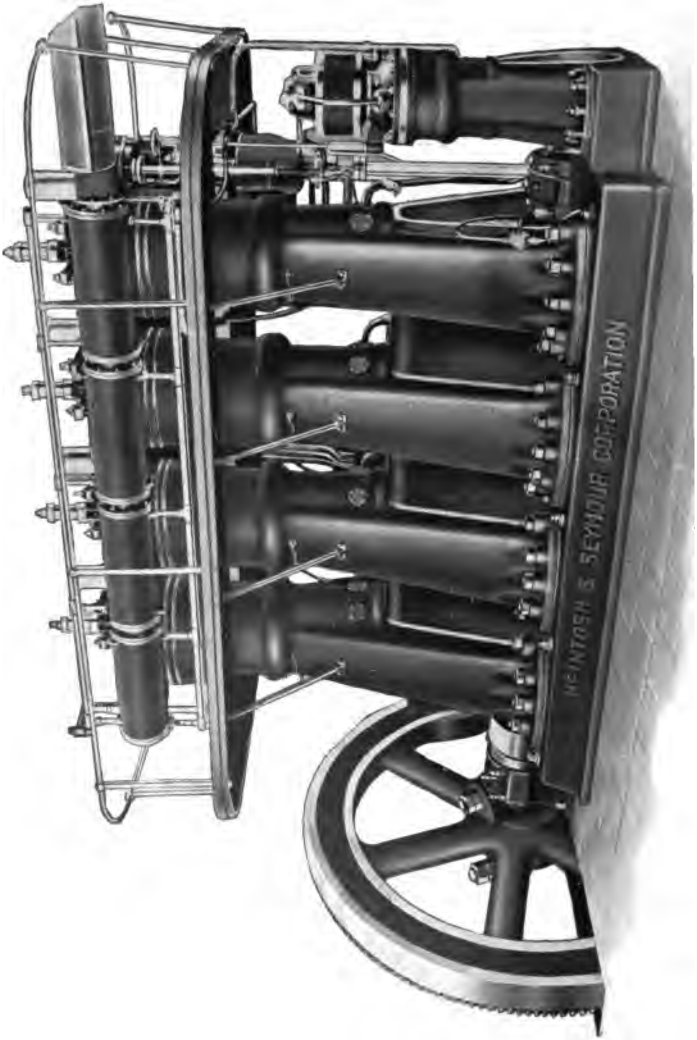
If they are not equal the cylinder which is working improperly should be examined both with regard to valves, fuel inlet and injection air.

WHEN STOPPING THE ENGINE which is effected by shutting off the fuel supply, the injection air receiver valve should be tightly shut to prevent leakage. The water supply should be also gradually reduced and its flow stopped, and if in cold climates and there is a possibility of freezing all jackets and pipes should be drained. The lubricators must also be shut off. The cooling air coils between the high pressure air and intermediate compressor should be blown out as well as the injection pipe.

CHAPTER VI

VARIOUS TYPES OF ENGINES

DIESEL POLAR ENGINE: The Diesel engine as built by the A. B. Diesels Motorer Co., Stockholm, Sweden, is shown in the 4 cycle type at Fig. 30. It is of the four cylinder type 550 B.H.P. In this size, the multi-stage high pressure air compressor is operated directly from the disc-crank placed on the end of the main crankshaft. In the smaller size engines from 25 to 200 HP., the air compressor is operated from the connecting rod by rocker arm as shown in Fig. 31. The A frame in the construction of this type is carried up to the upper part of the cylinder which is cast in one piece. A separate cylinder liner being inserted and the cylinder cover being bolted to the top of the casting. The bed plate cast in one piece for 3 cylinder engine showing the construction of main bearings is illustrated at Fig. 32. The cylinder head water cooled and which contains air inlet and exhaust valves, fuel inlet and starting air valves, and the arrangement of valve levers, cams and horizontal camshaft are shown at Figs. 33 and 34. The camshaft is operated by skew gears running in oil from a vertical shaft which is actuated from the crankshaft, also by skew gears. As will be noted from Fig. 31 the trunk piston is slightly concaved at top which theoretically is the correct construction allowing the volume of gases to be greatest


McINTOSH & SEYMOUR CORPORATION

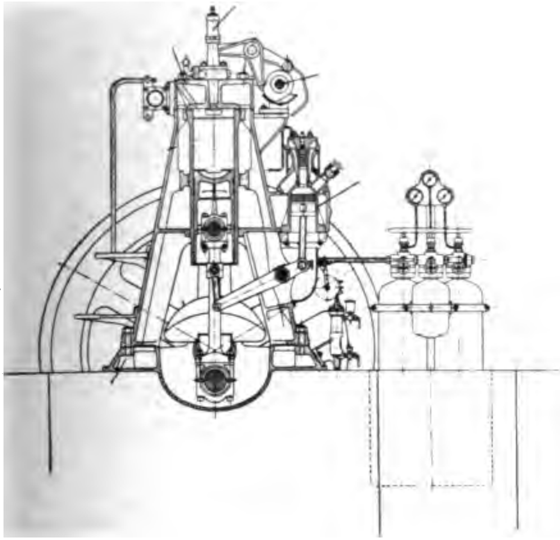


FIG. 31

FIG. 32

in the center. Lubrication of piston and cylinder is by means of force feed pump, which system is also applied to the multi-stage air compressor. The main bearings have ring oilers and the crankpins have each a centrifugal lubricator. The camshaft bearings have chain oilers.

FIG. 33

The governor mounted on the vertical shaft and the oil supply pump is shown at Fig. 35. With all engines the fuel supply pump, Fig. 36, has one plunger only, the fuel necessary for each cylinder being accurately distributed from the single pump plunger by a special distributing device. The fuel inlet valve is shown at

Fig. 34

Fig. 37 and the pulverizer peculiar to this make of engine, the function of which is to thoroughly pulverize (or mix with air) the fuel entering the cylin-

FIG. 35

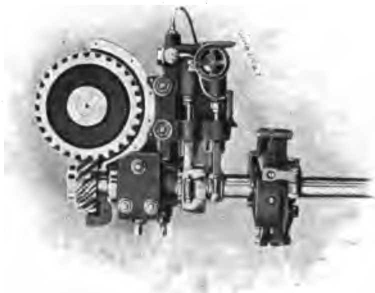


FIG. 36

der is shown at Fig. 37. In this device the fuel is forced through specially shaped passages giving the fuel and air a spiral motion thoroughly mingling parti-

Fig. 37

cles of fuel and air and producing a very efficient spray.

The following is a report of test made in Stockholm on a 65 HP. engine:

The two cycle marine Diesel engine made by the same builders is shown at Fig. 38 and in section at Fig. 39, which illustrate engines from 60 to 400 B.H.P. In larger size engines, the trunk piston illustrated is replaced by crosshead and shorter piston. In this type the scavenging air enters the main cylinder not through scavenging valves in the cylinder head but through ports in the cylinder wall. As the piston moves downward exhaust ports are first uncovered followed by the uncovering by the piston of the air inlet ports. Thus fresh air enters the cylinder thoroughly scavenging and filling it with pure air. The scavenging air and high pressure injection air are furnished by multi-stage air compressors placed in line with main cylinders. An interesting feature peculiar to this engine is the method of reversing. Operated from the Main crankshaft is a two cylinder double acting combined air motor or pump. When the engine is in ordinary operation these act as air pumps delivering compressed air to receivers at approximately 10 to 12 atm., which after passing through reducing valves is partly used for scavenging purposes. When maneuvering is required the double acting air pumps referred to having the cranks set at 90° from each other become air motors and being operated by the compressed air reduced from 12 atm. to 5 atm. are used for reversing or starting, the fuel from the main

TABLE II.

REPORT OF TEST.

LOAD FACTOR	1/1	3/4	1/2	OVERLOAD
Duration of Test..........................	30	30	30	10
Average Net Load Noted.....................	150 kg	110,5	73,5	182 kg
Lbs=.....	330,69	2436	162,04	401,24
Average Speed in Rev. Per Minute...........	299	302	305	297
Effective Work, B. H. P....................	64,8	48,2	32,4	78
Consumption of Fuel Per 1/2 Hour..........	6,1 kg	4,67 kg	3,47 kg	......
Lbs.=...........	13,45	10,3	7,65	......
Consumption of Fuel Per Hour......	12,2 kg	9,34 kg	6,94 kg	......
Lbs.=.............	26,9	20,6	15,3	......
Consumption of Fuel Per Hour and B.H.P....	0,188 kg	0,194 kg	0,214 kg	......
Lbs.=...	0,414	0,427	0.472	......

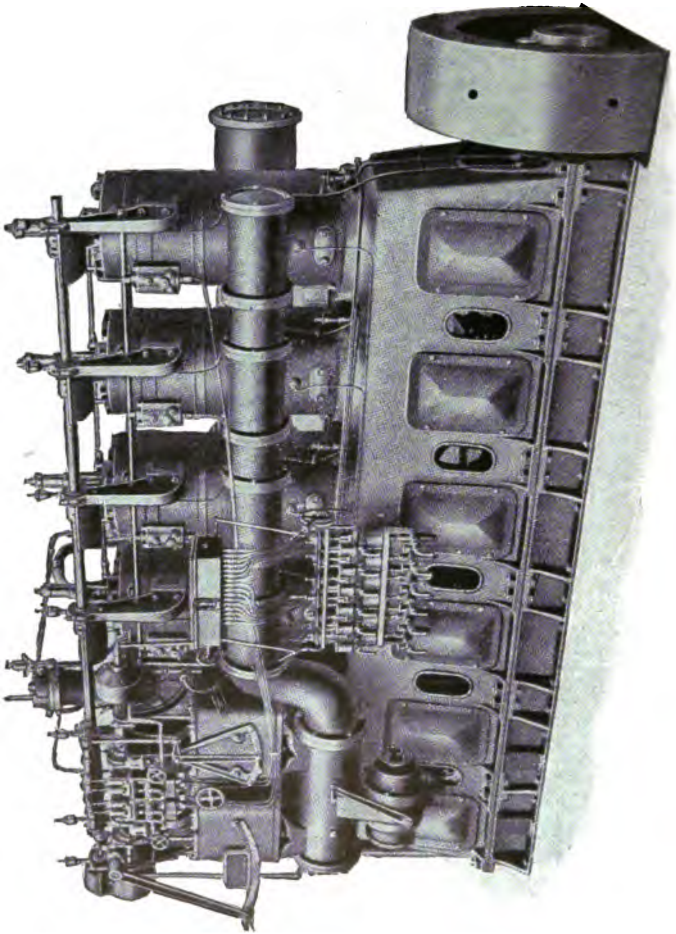


Fig. 38

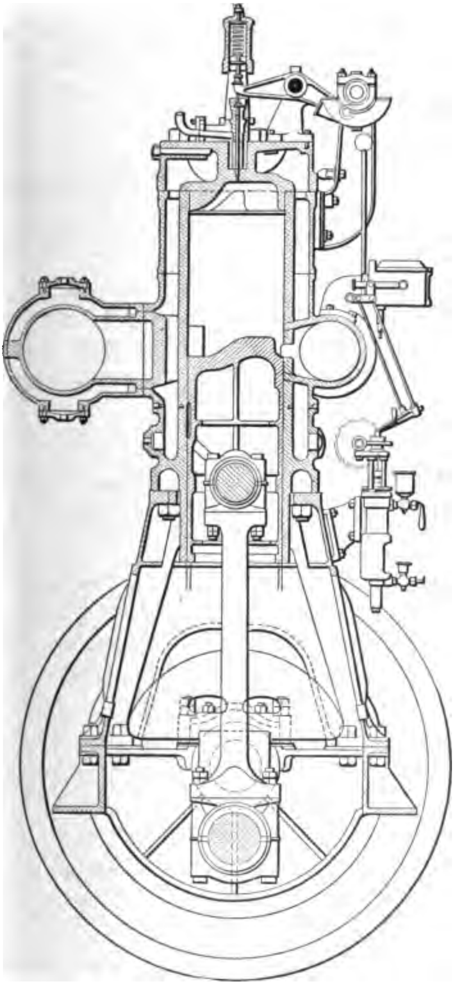


FIG. 39

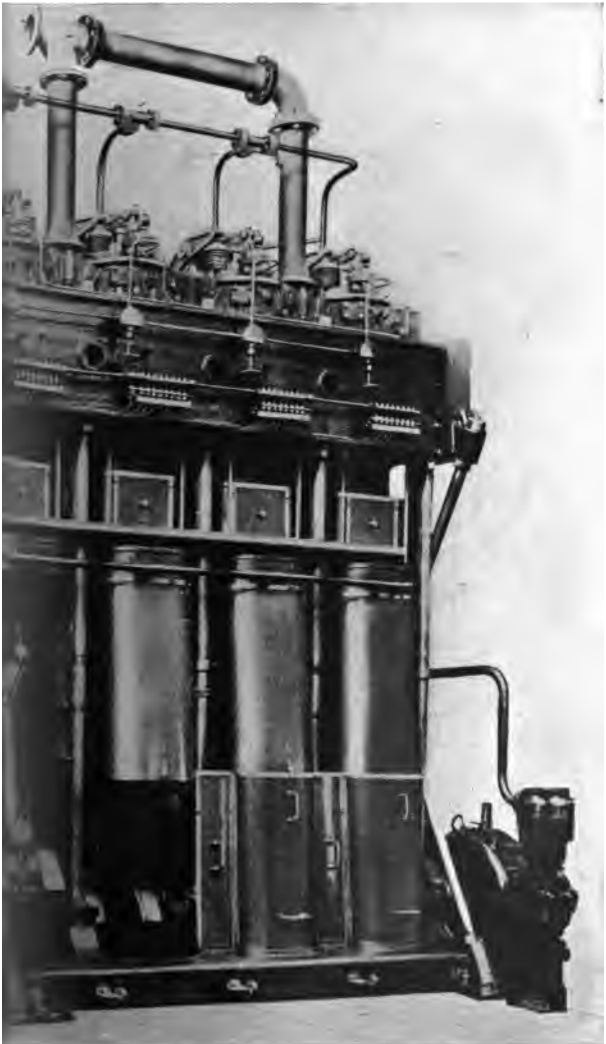
cylinders being cut off. The fuel valve is the only part requiring change for ahead or astern. This change is effected by means of a double set of cams for each cylinder. This method of reversing has the advantage of having no dead center, the air motor cranks being at 90° and the cooling effect found where air enters the motor cylinder is here avoided.

THE WERKSPOOR DIESEL ENGINE built by the Werkspoor Company, Amsterdam, Holland, is shown in Figs. 40 to 44. Fig. 40 shows 300 H.P. marine non-reversible type, with which reversing propeller is used. This company has completed over 33,000 H.P. and has in course of construction nearly as much of marine Diesel Engines which have been built in sizes from 1500 indicated H.P. down; Fig. 41 shows an engine of that capacity being of the 4 cycle type and equipped with 6 single acting cylinders each 22 in. diameter by 39½ in. stroke operating at 110 R.P.M. The total weight of the engine is approximately 120 tons complete.

An engine of this description in the ship "Juno" developed 1460 indicated H.P. at 115 R.P.M. and allowing 78% mechanical efficiency, this is equivalent to 1140 B.H.P. The M.E.P. of indicator card was 105 lbs. per square inch, the fuel used was 0.94 specific gravity. The fuel consumption being under 0.45 lb. per B.H.P. hour. In actual sea-going service the engines average 0.3 lb. per I.H.P. hour.

Two features peculiar to this engine are the valve motion and the open crank case construction illus-

PLATE I



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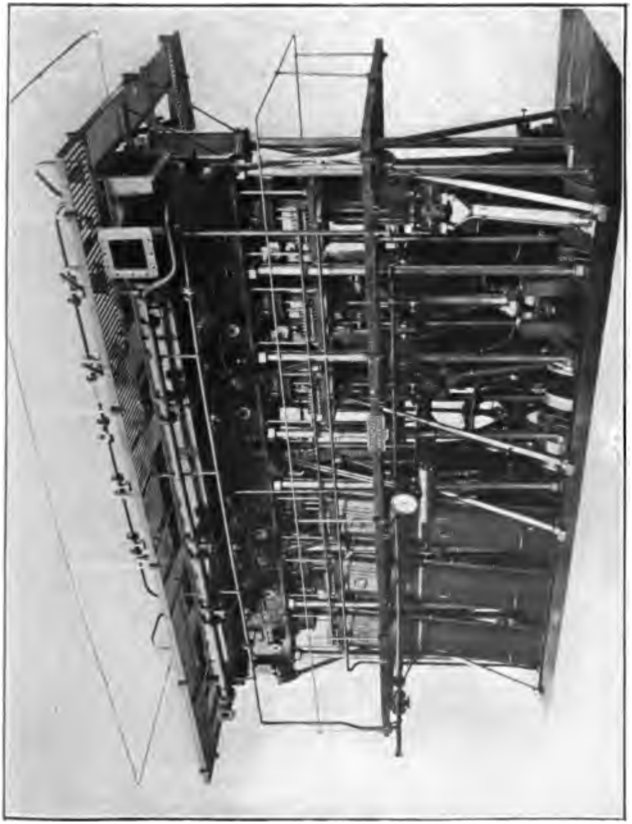


Fig. 41

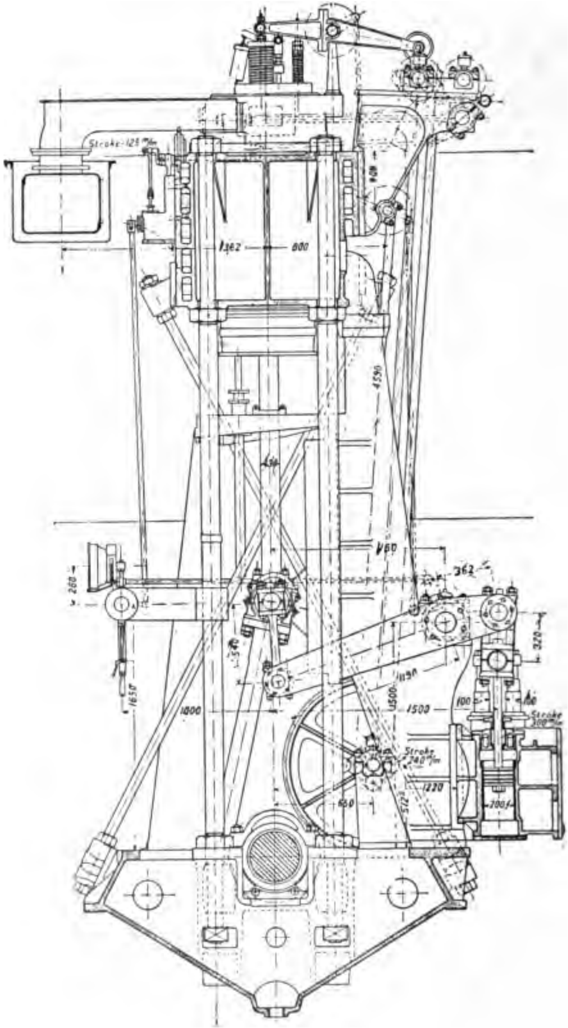

Stroke-125 m/m
1362
800
4590
1160
362
320
1650
1000
1190
1500
1500
100
100
Stroke 300 m/m
Stroke 240 m/m
1220
650
5123
200

FIG. 42

trated in Fig. 41, the cylinders being mounted on steel columns, this construction is made rigid by the use of steel cross-stays as shown in Fig. 42 and also by the cast-iron frames at the back of the engine. Detachable steel-plate doors are fitted to the steel columns to prevent the splashing of lubricating oil from the crankcase. Fig. 43*a* shows the method of detaching the lower part of the cylinder, and is interesting, as it enables any piston to be removed in less than one hour, which is most important from a marine engineer's point of view, seeing that it avoids removing the valve gear and cylinder head.

Crosshead, guides and reciprocating parts are shown in Fig. 42. The piston is water-cooled, the cooling medium being carried to the piston by clearance tubes and a pressure jet.

High pressure compressed air for injection purposes is furnished by three-stage air compressor which requires with this size engine approximately 100 H.P. This compressor is operated by means of a link and beam lever from the crosshead at the back of the engine. The cooling water and bilge pump are operated in a like manner from the forward cylinders. The after cast-iron column is used as an inter-cooler for the compressed air, the air circulating through water-cooled coils in it.

Lubrication is entirely by sight-feed lubricators. Lubricating oil escaping is conducted to a central sump whence it is pumped through a filter and cooler returning by gravity to the numerous lubricators.

The valve motion of this type of engine as previ-

ously stated is one of its peculiar features. It consists first of a small crank shaft operated by two to one reduction spur-gearing from the main crank shaft, see fig. 43. On this shaft is a four throw crank. By means of four long hollow girder-shaped rods the

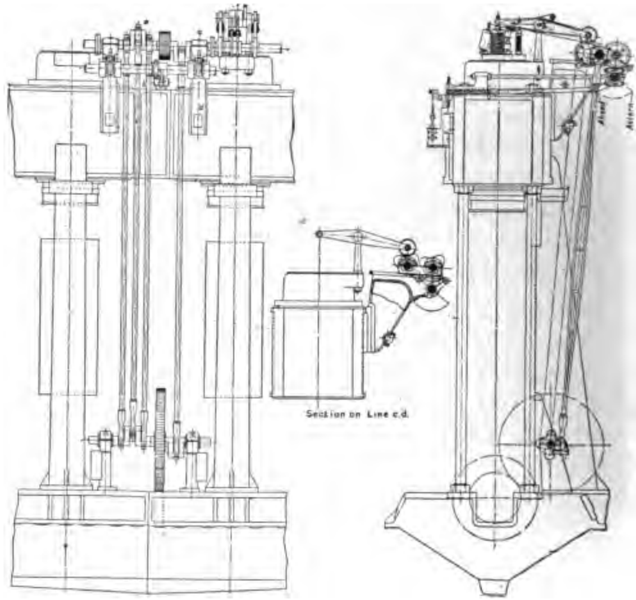


FIG. 43

motion of these cranks is transmitted to similar crank throws on the "ahead" cam shaft mounted on brackets on the cylinder head. Each crank is set at right-angles to the others and thus continuous rotary motion is obtained without putting any rod in compression.

There are two cam shafts, one for "ahead" and the

other for "astern" movement, the latter is driven by means of spur-gearing from the former and both shafts are mounted on sliding saddles. When maneuvering and it is required to reverse the engine then these saddles are slid to or fro. In this way the "ahead" or the "astern" cams are brought under the rollers of the valve rockers, which are eccentrically

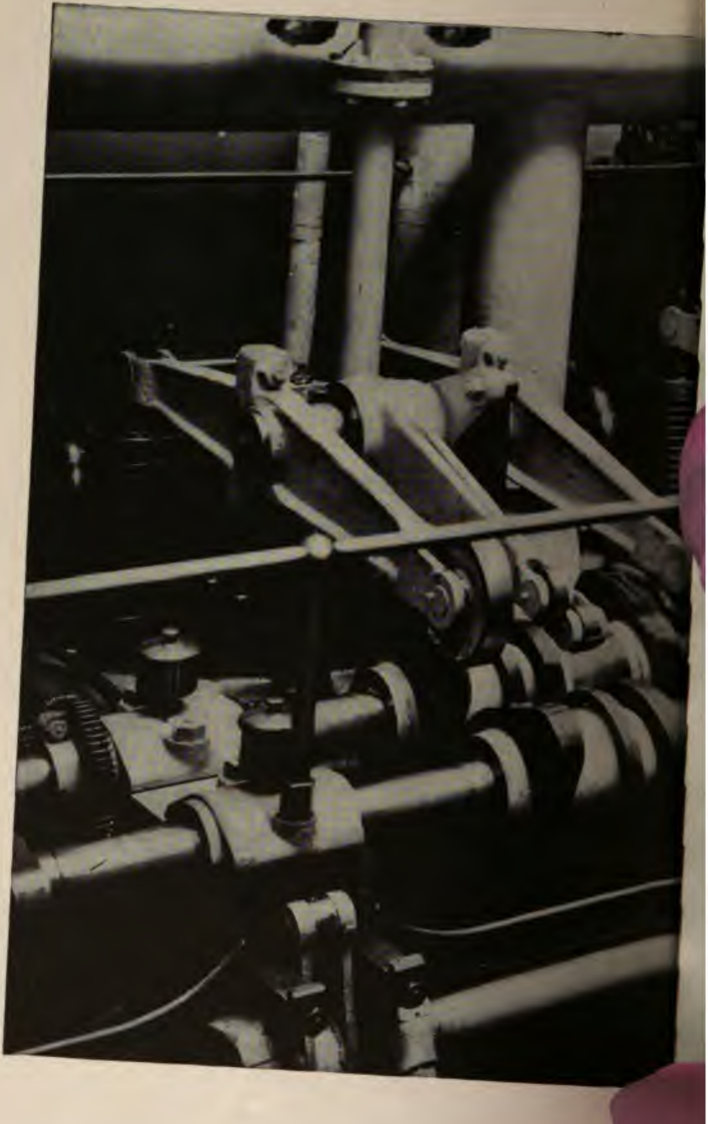
FIG. 43A

mounted so that the rollers can clear the cams when shifting the cam shafts. The rollers are of large diameter, so that the angle of the face of each roller when it meets its cam is not steep, which results in smooth and silent operation. As the four coupling rods' movement requires it the brasses are made with play and clear the pins when in the center or mid

gear position; they come again into contact when the shaft is in running position for either "ahead" or "astern." Fig 44 shows the arrangement of both cam shafts and the details of the valve motion.

The engine can be started with only 250 lbs. of compressed air. This is made possible by the use of air relief-valves fitted to each cylinder and also by choking the air inlet to the cylinders. The air starting cam has two lifts, one to allow the valve being slightly opened, which is used when the engine is hot, the second or larger lift is used when the engine is cold and requires more air.

GUSTO-DIESEL ENGINE—This interesting type of Diesel engine, built by N. V. Werf Gusto, firm previously known as the A. F. Smulders Co., of Schiedam, Holland, is shown in section at Fig. 45. It is of twin cylinder construction rated at 200 B.H.P. Each cylinder is 16 17/32″ (420 mm.) diameter, and 26 25/32″ (690 mm.) stroke. It operates at 210 revolutions per minute. The bed plate is cast in one piece. Each cylinder and liner together with the cylinder head is but one casting. The cylinder is supported on separate cast iron standards and through bolts are used to take the strain, these pass from the bed plate to bosses cast on the cylinders at the four angles, as shown in the illustration. This arrangement forms both a simple and rigid construction. The bed plate is made deep enough so that the piston can be withdrawn from the lower side. The crankcase is enclosed with sheet steel doors or covers tightly fitted and easily removed. The



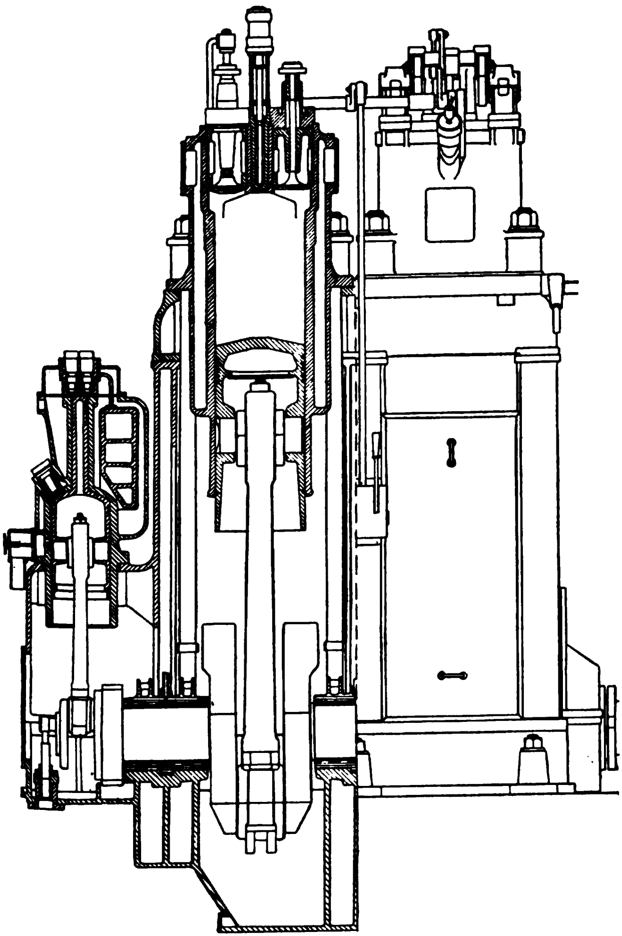


FIG. 45

trunk pistons are water cooled, the cooling medium is introduced to the upper part of the piston through a system of telescopic pipes. The supply of water is furnished not by a pump but by gravitation from a supply tank placed about 8 ft. above the level of the piston when in its upper position. The cooling water necessary for the main cylinder and air compressor cylinders is also supplied from this tanks by gravitation.

The air and exhaust valves are each placed in separate housings inserted in the cylinder head, as shown in the sectional view. The exhaust valve is hollow and is air cooled. The upper part of the exhaust valve stem is slotted and is fitted with a sliding joint arranged to allow the air to enter and leave the hollow stem and valve. The air starting valve is placed in the front part of the head at an angle of 45°. The fuel valve or sprayer is placed in the center of the cylinder head between the air inlet and exhaust valves. The valve motion consists of rocker arms pivotted on a center shaft. Each rocker is overhung, thus facilitating removal when it is necessary to withdraw the air or exhaust valves. The rocker arms are operated by cams placed on the horizontal camshaft at the upper part of the cylinder. The fuel is delivered to the sprayer by an independent pump for each cylinder; these fuel pumps are actuated by one eccentric attached to the vertical intermediate shaft. Force-feed lubrication for the cylinder is furnished by a pump also actuated from the vertical shaft. The centrifugal governor is placed at the top of the vertical shaft; it

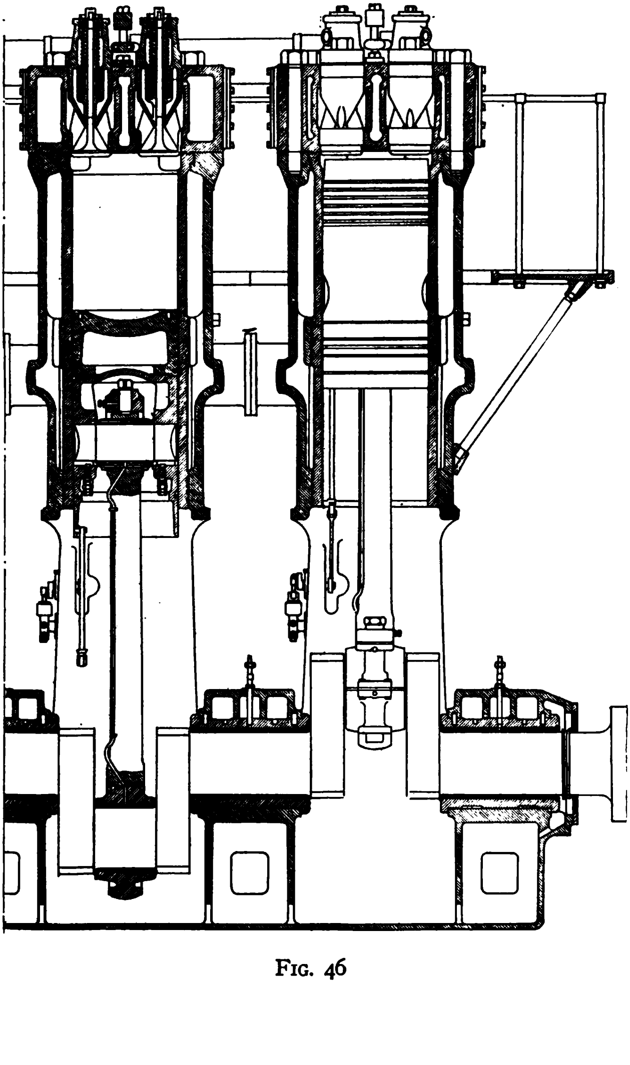


FIG. 46

regulates the amount of fuel passing to the sprayer by acting on the suction valves of the pump in the ordinary manner, that is, holding the suction valve open when required and thus reducing the amount of fuel passing it.

The single-acting two-stage air compressor has both cylinders cast in one piece or nearly so—the head is a separate steel casting. The compressed air is cooled after emerging from the first stage compressor and is again cooled after passing from the high pressure stage. The compressor pistons are actuated from an overhung crank placed on the end of the crankshaft.

THE WILLANS DIESEL ENGINE—The Diesel engine as made by Messrs. Willans & Robinson, of Rugby, England, and the Dow Pump and Diesel Engine Co., is shown in section at Fig. 46 and Fig. 47.

These engines are built in sizes of 50 H.P. to 960 H.P., being either single or multi-cylinder type. The illustration shows a 4-cylinder 640 H.P. They are all made of the vertical type and as will be seen from the illustration, are of the "A" frame construction; the cylinder casing is cast in one piece with the frame, separate cylinder liners being inserted into the casting. The Willans Diesel engines are of the excellent design and workmanship in all details for which this firm is justly famous.

One of the features peculiar to this type is the cylinder head. It is made with two cored recesses only, one for the air inlet valve and housing, and the other for the exhaust valve and housing. The fuel inlet valve

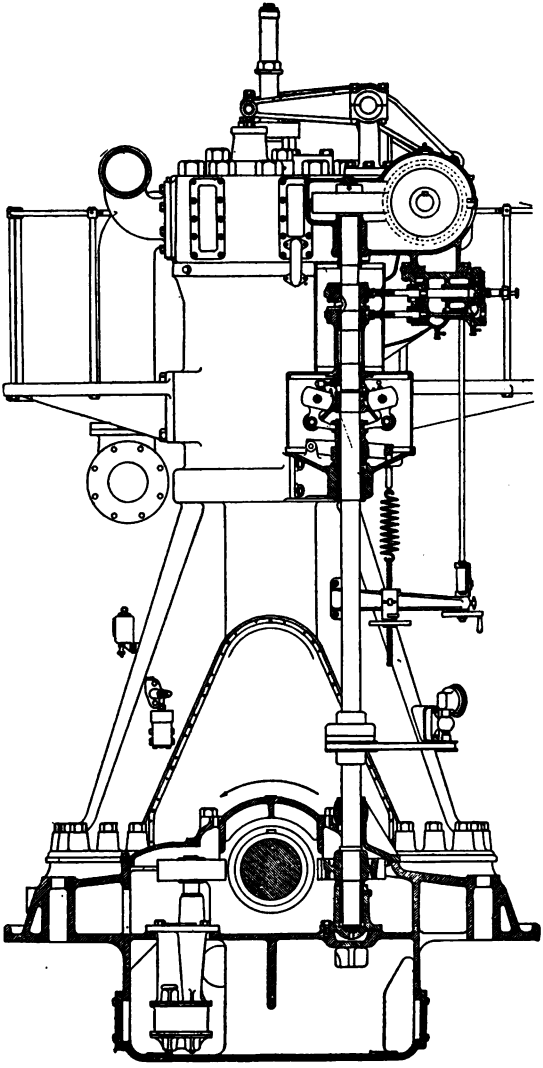


FIG. 47

or sprayer is inserted in a tube. This tube has different diameters at each end and is pressed into its place between the lower and upper facings of the walls of the cylinder head and is held rigidly in place by suitable bolts. No recess is cored in the cylinder head for the sprayer. In this way the shape of the casting is simplified and unequal masses of metal are eliminated and trouble frequently experienced from cooling or shrinkage strains of the casting, is done away

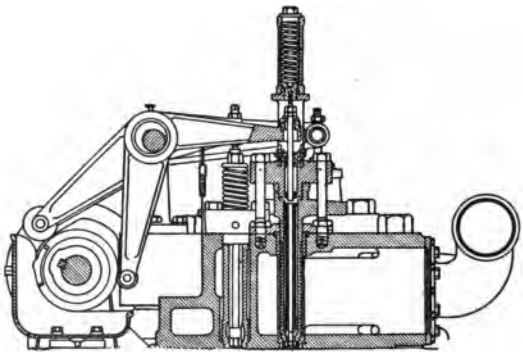


FIG. 47A

with. A small space, about ¼″ wide, is left for the circulation of the cooling water around the tube into which the sprayer or fuel inlet valve is fitted; it is shown in section at Fig. 47*a*.

The piston is oil cooled and as may be seen from the illustration, is made in two parts, the upper part which is hollow for cooling purposes is connected to the lower part by bolts. A special form of telescopic

piping conducts the cooling medium to and from the upper part of the piston.

Only one fuel supply pump is furnished for supplying the four cylinders, the fuel being conducted to each sprayer through a fuel distributor. The makers claim an advantage for this arrangement, stating that with individual fuel pumps for each cylinder, if the valves become worn then the distribution of fuel is not equalized as it is with their arrangement, and that with one pump only and their special fuel distributor the same amount of fuel is delivered to each cylinder and thus it is impossible to overload one cylinder while another is doing less than its share of work.

The governor varies the speed of the engine by operating on a relief valve placed in the fuel pump cylinder. In this way the lift of the valve is varied, thus regulating the amount of fuel delivered to each sprayer.

Lubrication of the main bearings is effected by the lubricant passing to the bearings through the hollow crankshaft. Each of the connecting rod bearings are also lubricated in this way, the gudgeon pin obtaining its supply through the oil pipe placed by the side and attached to the connecting rod leading from the large end or crankshaft bearing to it.

M. A. N. Horizontal Diesel Engines—For land installations the Maschinenfabrik Augsburg-Nürnberg since about 1910 have been manufacturing the horizontal type of Diesel engines in addition to the vertical types which they have built so largely principally

for land installation as well as for marine purposes. These horizontal engines are built in both single-acting and double-acting type 4 cycle as well as 2 cycle single-acting. The single-acting type are built in single, twin and four cylinder design, while the 4 cycle is also built tandem double-acting.

Four Cycle Single Acting—The 4 cycle single acting single cylinder 150 H.P. engine operating at 165 r.p.m. is shown in section at Fig. 48, which illustration clearly shows the arrangement of the air and exhaust valves and fuel injection valve. The separate cylinder liner is supported from the casing at two intermediate points. The trunk piston is made in two parts. The head or part of the piston exposed to the heat of combustion is composed of a mixture of cast iron best suited to withstand the heat. The trunk or main part of the piston is a mixture of hard close-grained iron, most suitable to withstand wear and allow of good lubrication. This design of piston is fitted with two adjustable strips placed in recesses on its upper side which are furnished to allow of adjustment when it is necessary, thus taking up the wear of the piston. For this arrangement the makers claim all the advantages of a crosshead and guide without requiring the space necessary for the crosshead design and without the additional cost in manufacture entailed by the use of the crosshead. The same design of piston is used on their vertical land engines also.

The valve motion consists of one eccentric placed on the camshaft with suitable rods between it and the

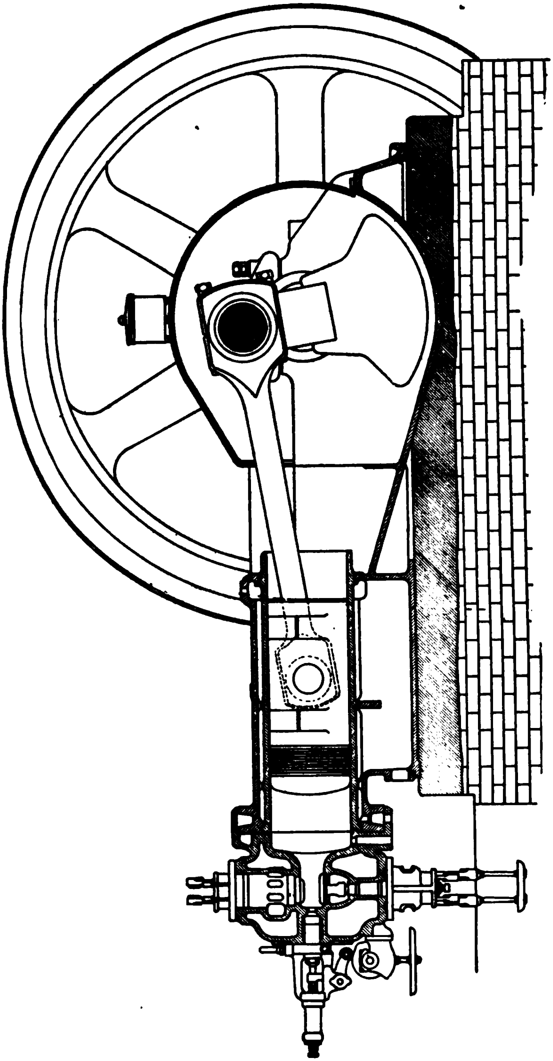


FIG. 48

rocker arms of both air inlet and exhaust valves. This is a similar arrangement to that shown with the 2 cycle valve motion at Fig. 50. The side shaft is geared again to a crossshaft placed behind the cylinder which is furnished to operate the fuel injection valve. The air compressor for furnishing high pressure injection air is driven from the outer end of the crankshaft and is attached to the bedplate in the single cylinder engine; in the multi-cylinder type it is placed separately on the foundation.

Lubrication is supplied to all bearings by a feed pump placed in the compressor crankcase which elevates the lubricant to a receptable placed above the crank cover, whence it flows to each bearing. This oil is filtered and used continuously. The piston is lubricated by a force feed pump in the usual way.

In the double cylinder engine of this type the frame is cast in one piece; the valve motion consisting of eccentric and push rods is similar to that already described, only that they are extended to the valves of the second cylinder, and the crossshaft operating the fuel injection valve is also extended (see Fig. 53).

In the 4 cylinder engine of the 4 cycle type two sets of two cylinders are placed side by side with the flywheel in the centre.

Two Cycle Single Acting Engine—The 500 H.P. twin cylinder 2 cycle type is shown in section at Fig. 49. A back view illustrating the valve motion crossshaft and governor is shown at Fig. 50. These are built in twin cylinder design of 500 H.P. and upwards.

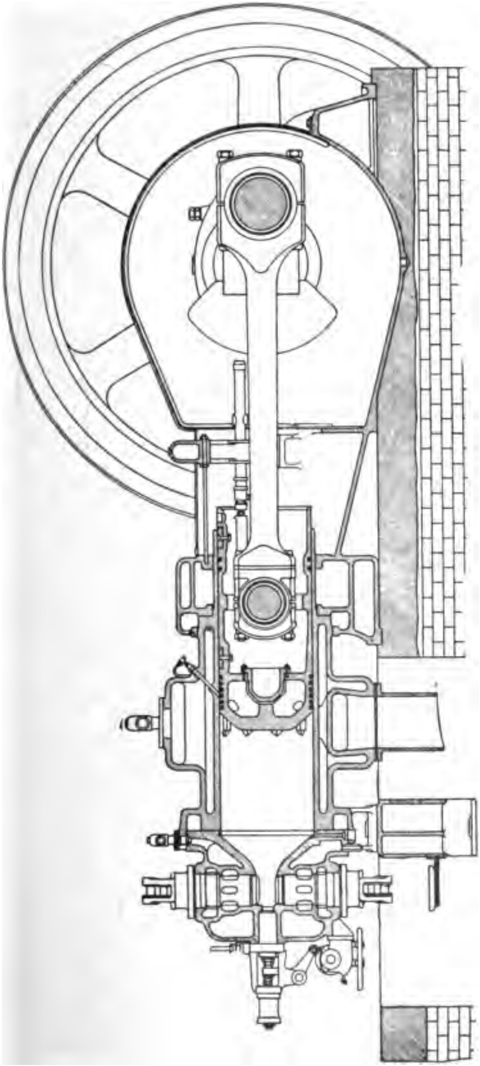


FIG. 49

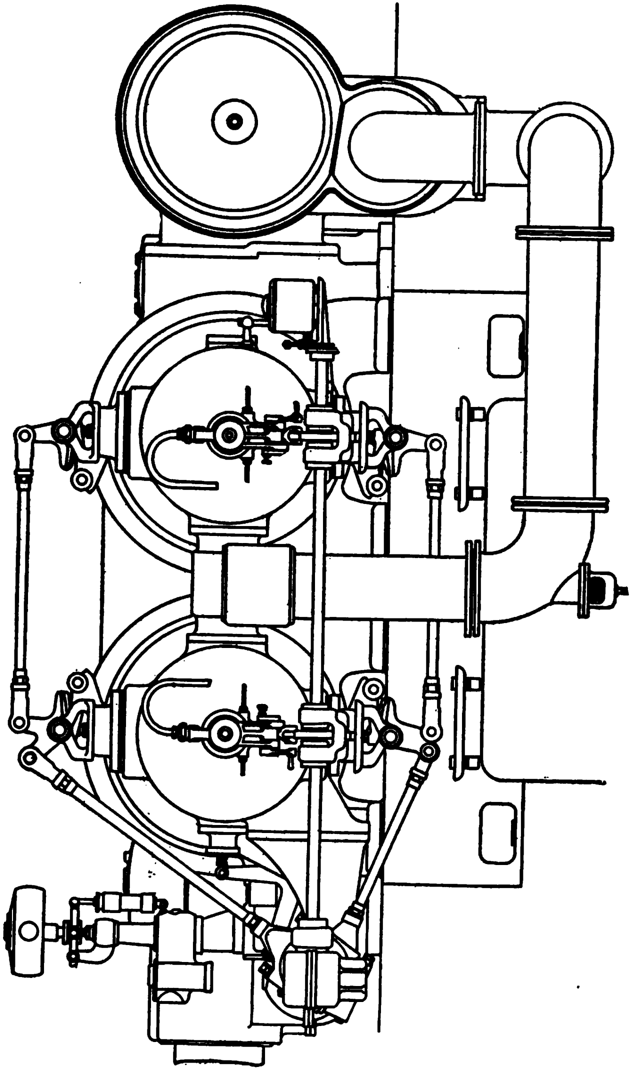


Fig. 50

The general arrangement is very similar to the 4 cycle type already referred to. The two scavenging air inlet valves are placed in the cylinder head. The exhaust gases are expelled through the ring of exhaust ports shown in the sectional view.

The piston is water cooled in all engines developing 175 H.P. or more in one cylinder. Telescopic tubes convey the cooling water to the piston in the ordinary way. The piston of the 2 cycle type is equipped with adjustable pieces fitted to it, as previously described for the 4 cycle engine.

The double acting scavenging air pump is placed in tandem with the high pressure compressor and is actuated from the crankshaft.

The twin cylinder 2 cycle type of engines are made in sizes of 500 H.P. operating at 165 r.p.m., also 750 H.P. operating at the same speed and 1,000 H.P. running at 150 R.P.M. A 1,000 H.P. double acting 2 cycle single crank engine has been built by these makers and is installed in their own power house.

M. A. N. 4 Cycle Horizontal Double-Acting Diesel Engine is shown at Fig. 51 and Fig. 52. In 1908 this company built the first engine of this type developing 600 H.P. Diesel engines of this design have since been built of the twin tandem construction up to 4,000 H.P., that is 500 H.P. in each combustion chamber. The general arrangement of the construction of these (the largest Diesel engines built) is shown by the different illustrations. Fig. 51 shows a tandem double-acting 4 cycle 800 H.P. operating at

136 R.P.M., having cylinders 650 mm. (25 9/16″) diameter and 900 mm. (35 7/16″) stroke.

Particular attention has been given to the cooling system. The space for cooling water around the cyl-

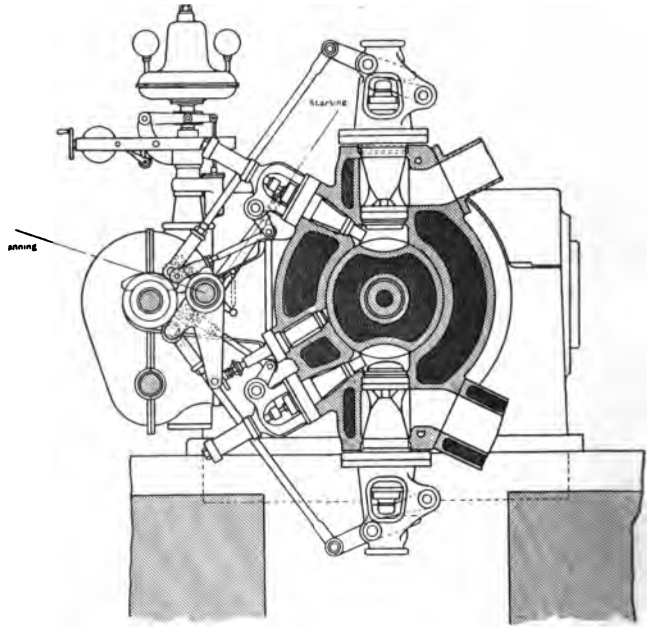
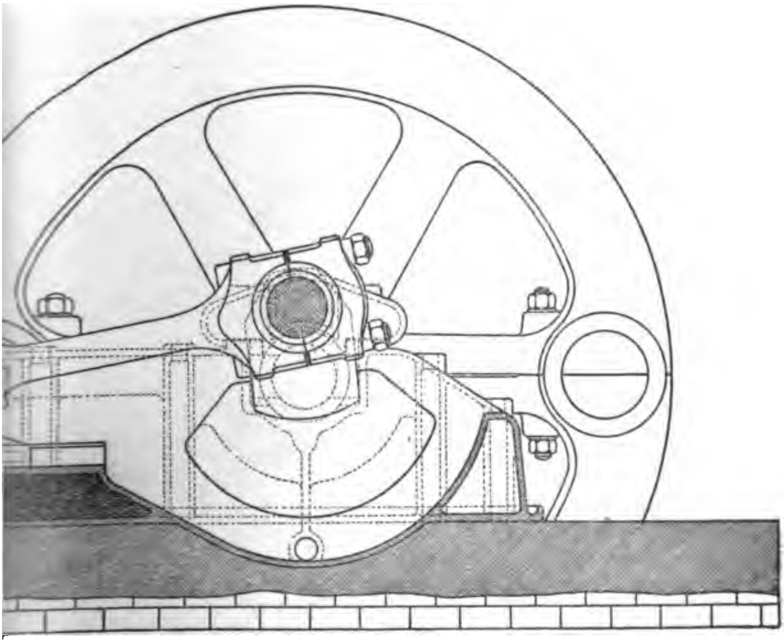


FIG. 52

inders has been made extra large so as to effectively carry off the heat developed with combustion each side of the piston. The pistons are each water cooled as shown, with the cooling medium circulated through the hollow piston rod, which first enters through the swivel-jointed piping attached to the upper part of the

PLATE III



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central slipper shown in Fig. 51. The cooling water enters under a pressure of about 90 lbs. so as to insure rapid and proper circulation. The exhaust valves themselves are not cooled, but the valve housing is arranged with cooling spaces in it.

The bedplate and cylinders are in six sections, viz.: The bedplate—two cylinders, the trunk and pedestal connecting the cylinders, the rear guide and the pedestal for supporting it. In each cylinder are two air inlet valves and two exhaust valves, one of each for each combustion space; the air inlet valves are placed above and the exhaust valves are situated at the lower part of the combustion space. The camshaft is actuated by spur gearing from the forward lower shaft which is geared again to the crankshaft. Arrangement of the valve motion levers and other mechanism is shown in Fig. 52.

Each combustion space is fitted with two fuel inlet valves placed about 120° apart in the cylinder. They are controlled by separate cams so timed that the injection of fuel from each is simultaneous. A separate lay shaft is furnished for each cylinder, placed behind the camshaft on which the fuel injector motion rods, have their radius links eccentrically mounted. Air for starting purposes is furnished through valves placed in each end of the forward cylinder only; when starting the fuel inlet is cut out on the forward cylinder, but is allowed to enter the rear cylinders. The sectional view shows how the cylinder covers act as housings for the piston rod stuffing boxes. The weight of the piston and rods is borne entirely by the crosshead guide as

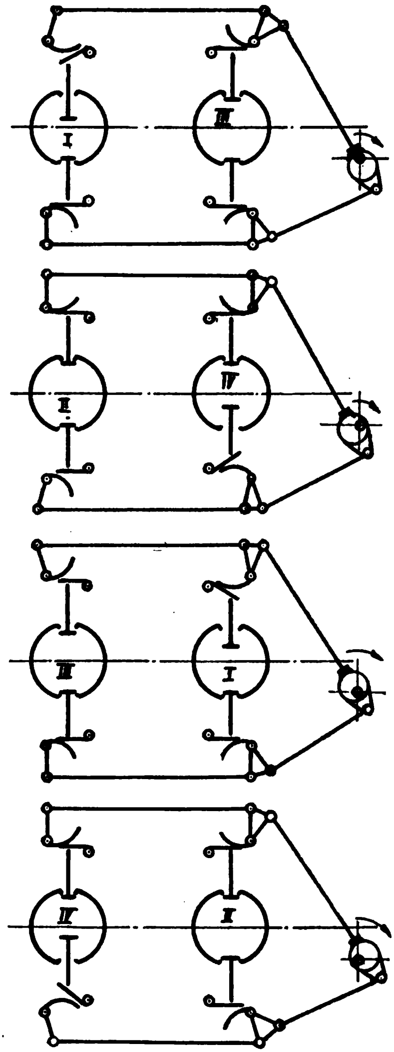


FIG. 53

well as by the intermediate and the rear slipper guides. The function of the stuffing boxes is thus only to prevent leakage between the gland and the moving rod; they bear no weight and are built up of a number of sections containing triple rings with provision for lubrication in the centre. All crosshead and slipper guides are provided with force feed lubrication; the main bearing and crosshead pin are all lubricated in the same manner. A fuel pump is furnished for each cylinder end. These pumps are placed in the centre of the engine separately operated from the camshaft by eccentrics. The governing is effected in the ordinary manner by varying the duration of opening of the pump suction valve.

Fig. 53 illustrates diagrammatically the operation of the valve motion of the four cycle type and similar to that shown in Fig. 50. I is the air suction. II the compression. III expansion, and IV exhaust strokes.

THE JUNKERS OIL ENGINE.—Briefly described, this engine operates on the two cycle plan, it consists of motor cylinder of greater length than other engines in which operate two pistons. The piston nearer the crankshaft is connected to its crank in the ordinary way, the piston farthest from the crankshaft moves in the opposite direction to that of the previously named piston, and is attached at its back end to side-rods supported on each side of the cylinder, which are actuated through connecting rods from cranks each side of the main crank. Thus a three throw crank is required, the two outside cranks being in line with

each other, and are set at 180° from the main centre crank. In the motor cylinder walls are two sets of ports, one set for air inlet, the other for exhaust.

The method of operation is as follows: As combustion commences the pistons travel in opposite directions. Toward the end of the stroke the forward piston first uncovers the exhaust ports, then the back piston uncovers the air inlet ports, allowing pure air at a slight pressure to enter the cylinder and scavenge it thoroughly, on the backward stroke the pistons approach each other again performing compression, at the dead centre fuel is injected and expansion begins again. For marine service this engine is designed with one or two cylinders placed tandemwise and having four pistons in all.

Many advantages are claimed for this design, among which is the elimination of cylinder head and absence of strains through the cylinder, complete balance of the reciprocating parts, improved lubrication of the pistons and cylinders, high aggregate piston speed, ideal combustion space and decreased loss of heat to the cylinder water jackets.

THE A. E. G. DIESEL OIL ENGINE, which is an adaptation of the Junkers principle of design already described, is shown in section at Fig. 54 and Fig. 55. It has been made in various sizes from 15 H.P. twin cylinder type and also in the 4 cylinder construction up to 500 H.P. It operates in accordance with the method of operation of the Junkers type or Oechelhauser gas engine. Each cylinder has two pistons

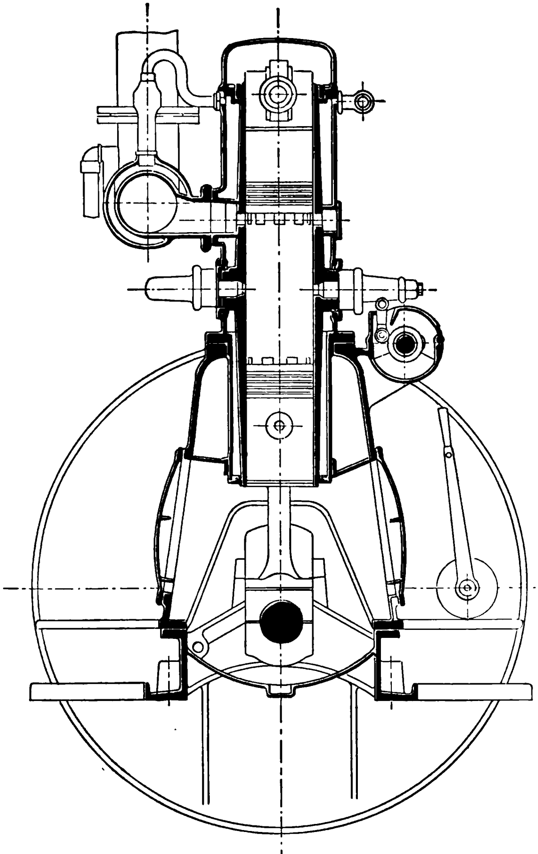


FIG. 54

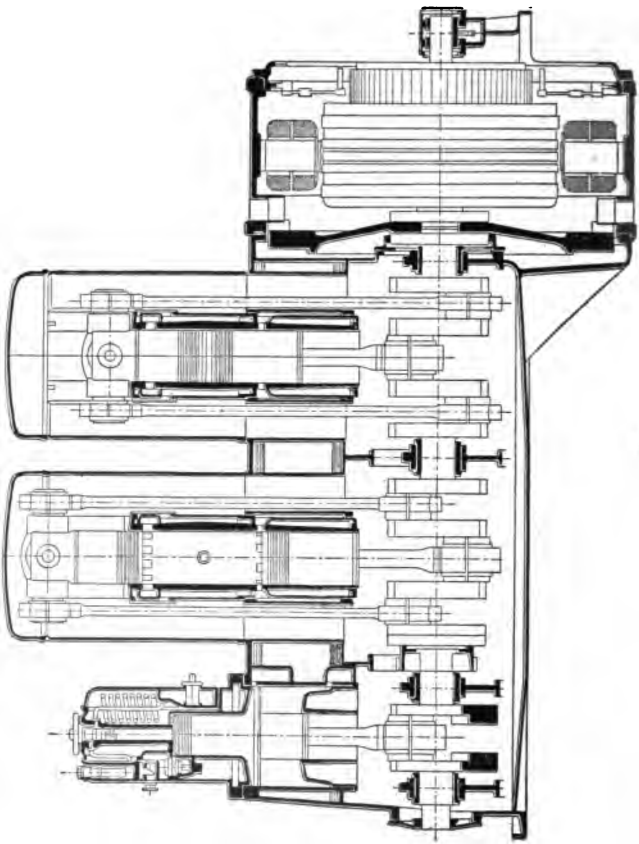


FIG. 55

which move simultaneously in opposite directions. The upper piston has a crosshead on its upper part with guides and long connecting rods, which are each attached to the two outward cranks placed at 180° from the centre crank. The high pressure air for injection with the fuel and the scavenging air at about 2 lbs. pressure is furnished by the two stage compressor and the air pump placed in tandem with it, and all actuated from the crank at the opposite end of the frame to the dynamo. A reservoir for the scavenging air is formed in the upper part of the crank case. The fuel injection valves and fuel supply pumps for each cylinder are operated from the horizontal camshaft which is actuated by worm gearing and vertical intermediate shaft from the crankshaft. The governor is placed on the end of the camshaft and regulates the supply of fuel by varying the length of the pump stroke. The fuel is injected into the combustion space between the pistons when closest to each other. Particular attention has been given to afford thorough and efficient lubrication of all moving parts, and oil-tight covers are placed over all these moving parts. The starting air valve is placed in the cylinder opposite to the injection valve. The pistons in the larger sizes are water cooled, and in the intermediate sizes they are oil cooled.

Figs. 56 to 60 left for future additions.

CHAPTER VII

VARIOUS TYPES OF ENGINES—Continued

THE MIRRLEES-DIESEL ENGINE built by Messrs. Mirrlees Bickerton & Day is shown at Figs. 61 to 65. These engines are made in various sizes up to 750 H.P., which is of the six cylinder type and runs at 200 R.P.M. They are of both the open crank case and the enclosed design.

The High speed closed crank case type with forced feed lubrication operating at a speed as high as 400 and 450—R.P.M.—is illustrated at Fig. 61, which shows the three cylinder 45 B.H.P. enclosed type direct connected to generator. Fig. 62 shows this design of engine as used for stationary purposes in cross section, while Fig. 63 shows in section the air inlet exhaust and fuel inlet valves, main bearings and other details of construction. The exhaust valve, housing, and valve lever are shown in Fig. 64. The lever is constructed as shown in order that the one half of it can be easily removed and thus facilitate the removal of the valve. The operation of the governor is shown diagrammatically at Fig. 65. The speed of the engine is varied as required by the action of the governor on the suction valve of the fuel supply pump. This valve is held from its seat by the governor during a part of the fuel pump plunger stroke, accordingly a varying amount of fuel is delivered to the sprayer according to the portion of the pump stroke that the suction

valve is allowed on its seat. It will be noted from the various illustrations that the air compressor is oper-

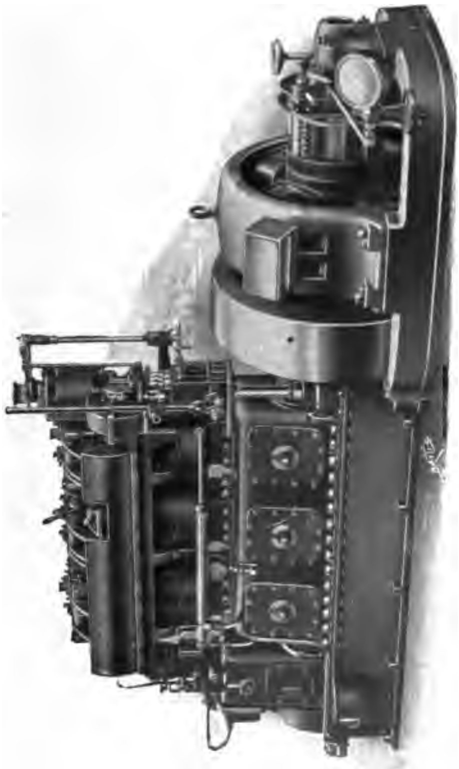


FIG. 61

ated directly from the crankshaft, which arrangement is now becoming standard practice. The fuel consumption of this type engine as guaranteed by the

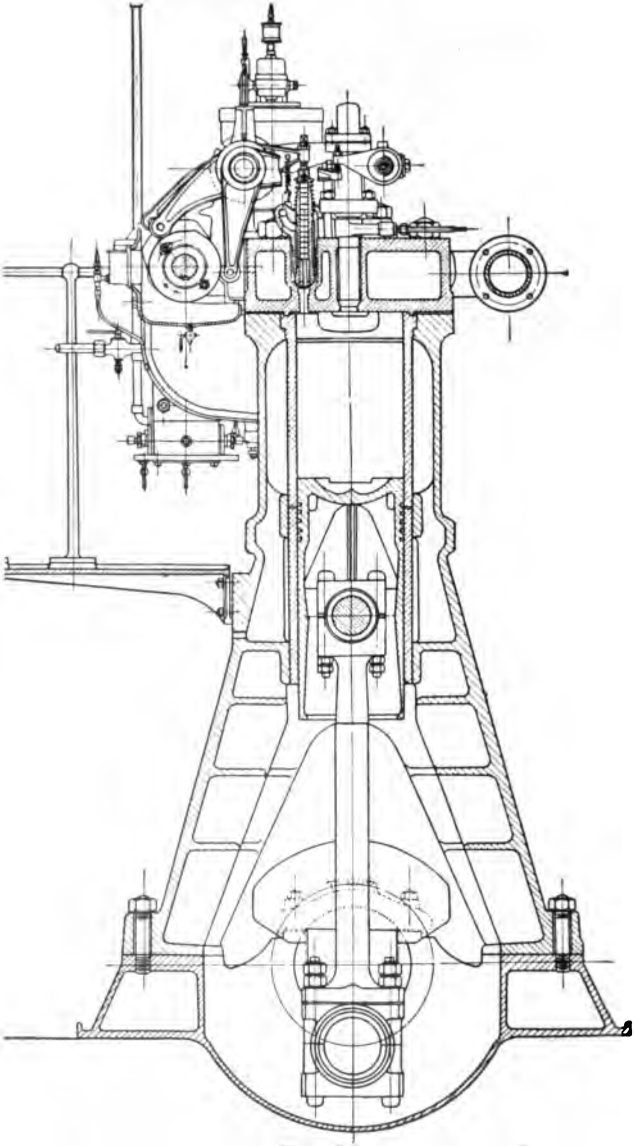


FIG. 62

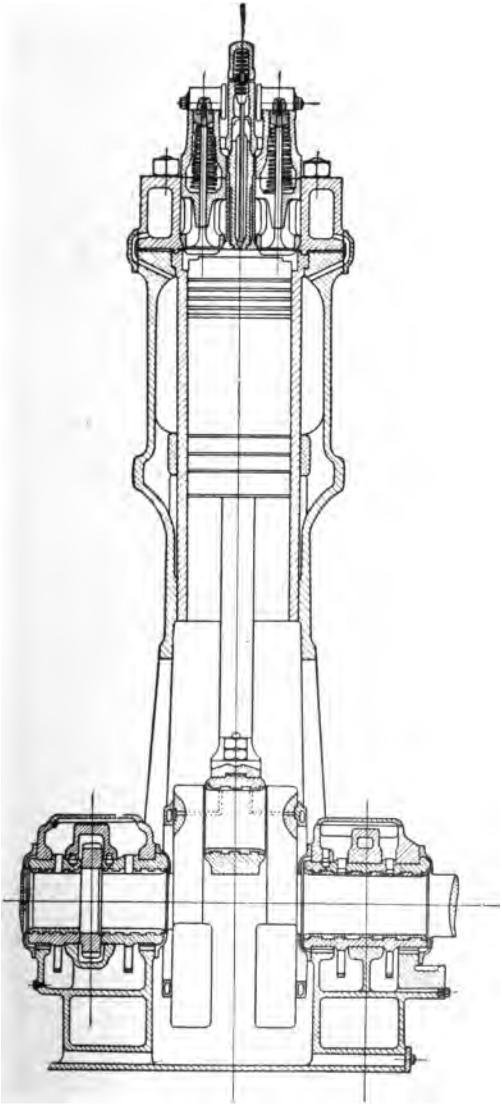


FIG. 63

makers is as follows: Full load .47 lb., three-quarter load .49 lb., half load .57 lb., one quarter load .76 lb.

Speedway Diesel Engines: Fig. 66 to 68 show sectional views of a 175 B.H.P. vertical two cycle

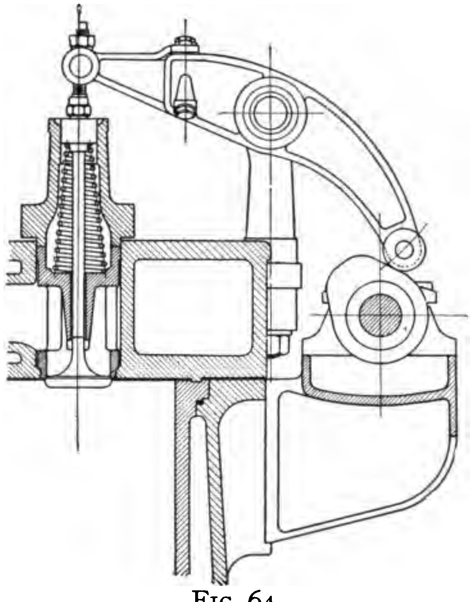


Fig. 64

Diesel engine, built by the Gas Engine & Power Company.

This type operates on the ordinary two cycle plan of operations, the air for scavenging purposes being compressed in a double acting air pump operated from the main crankshaft. The two stage air compressor

for furnishing high pressure air is operated by a beam lever from the connecting rod. The scavenging air enters the motor cylinders through scavenging valves placed in the cylinder head. The camshaft is placed

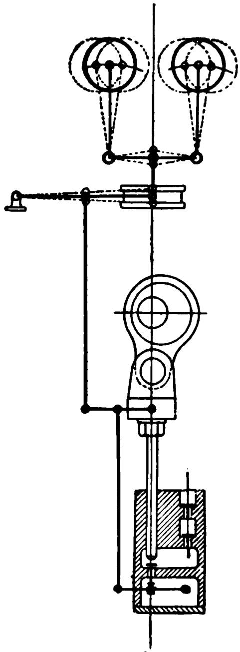


FIG. 65

parallel with the crankshaft at the lower part of the cylinders and is actuated directly from the crankshaft by gears, the fuel inlet and scavenging valves being operated by means of push rods and levers as shown in Fig. 68. The motor cylinder is 9″ diameter and 12″

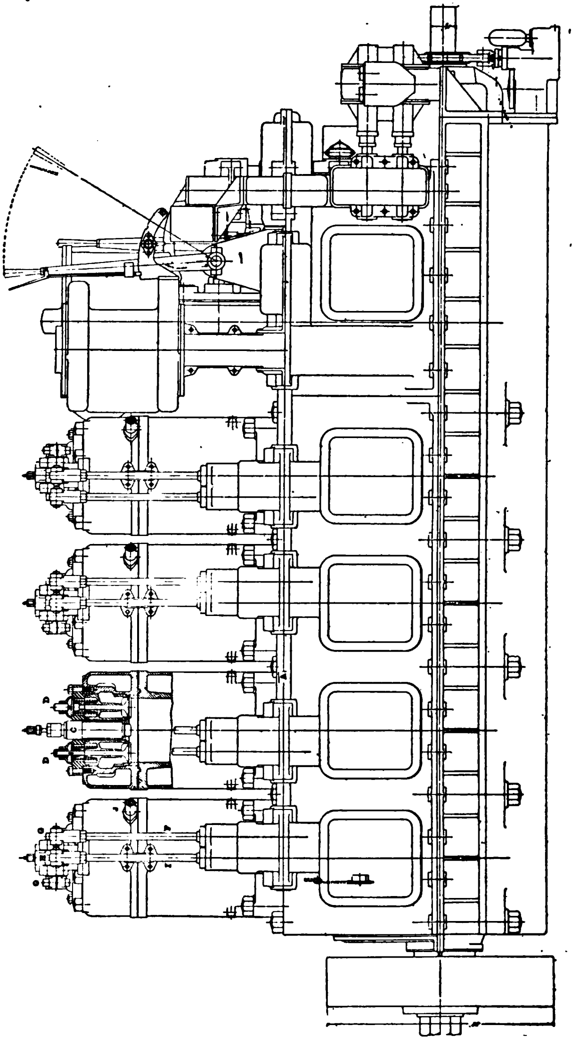


Fig. 66

stroke. The engine operates at 350 R.P.M. The two scavenging valves placed in each cylinder head are each $2\frac{3}{4}''$ diameter. The two rocking levers actuating these scavenging valves are worked by a single push rod. In the side of each cylinder head is an automatic

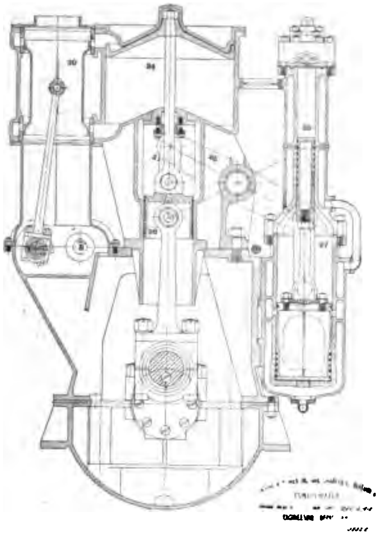


FIG. 67

relief valve set to blow off at 750 pounds per sq. in. Each fuel valve and pulverizer is contained in a detachable steel cage suitably bolted to the cylinder head. Each cylinder is equipped with its own fuel pump which is operated by eccentric from a small lay-shaft. Control of the fuel supply is effected by opening the

suction valves of the pump when it is necessary to cut off the fuel supply. The double acting scavenging air pump is 18½" diam. and 10" stroke. The high pressure air is intercooled between the two stages of compression. On the left of the scavenging pump is

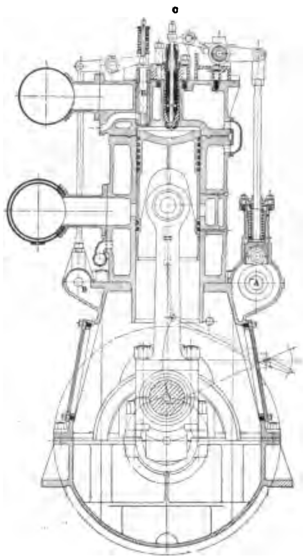


FIG. 68

shown a piston type valve which is utilized for controlling the air supply to the reservoir manifold located alongside the cylinders. It is operated by a crank and connecting rod from a lay-shaft driven by gearing from the camshaft that actuates the scavenging and fuel injection valves. This mechanical valve

is used instead of the automatic inlet valve and its use insures quiet operation. Lubrication of piston, wristpins, connecting rod, crankpin bearings and main-shaft bearing are all by pressure feed system, the lubricating oil being forced through the hollow crank-shaft by a pump driven from the afterend of the crankshaft. The crankcase is enclosed. The thrust is built with a roller bearing of large size. In the crankcases are fixed cast iron splash guards which prevent the connecting rod throwing excess of oil on to the cylinder walls.

Reversing of the engine requires all valve cams to be instantly placed in correct position for astern motion. To effect this—the scavenging valve and fuel inlet camshaft is fitted with a spiral sleeve, by which it may be turned axially at an angle of 30 deg. by means of a hand lever. The starting valve cams are moved longitudinally, thus bringing another set of cams into action under the push rods. To facilitate this movement these cams have tapered approaches.

Reversing is effected by the following movements:

1.—Fuel is shut off.
2.—Camshaft is shifted by hand lever.
3.—Compressed air is admitted to the cylinders which are on the upstroke.
4.—Admit fuel to cylinders.
5.—Shut off air supply. This requires about 12 seconds.

As will be noted from the illustration all the levers

are arranged close together on the forward end of the engine.

Daimler-Diesel High Speed Marine Engines, built by the Daimler Motoren Gesellschaft of Marienfelde, W. Berlin, are shown in section at Fig. 69 and Fig. 70. The 4 cycle engines as illustrated have been built in two sizes of cylinders, viz.: 160 mm. diameter (6 11/32″) and 230 mm. (9 1/16″) stroke operating at a speed of 560 r.p.m.; the larger size was 200 mm. (7⅞″) diameter with 230 mm. (9 1/16″) stroke and was operated at a speed of 530 r.p.m. The smaller engine developed 65 H.P. having four cylinders, and 100 H.P. with six cylinders. The larger size developed 100 H.P. with 4 cylinders and 160 H.P. with 6 cylinders. The illustrations show the 4 cylinder 100 H.P. size. Being of very compact and light construction it weighed about 62 lbs. per B.H.P. complete with piping and other accessories.

The crankcase is made of bronze, being cast in one piece with the cylinders and are made in units each comprising two frames and two cylinders. Each cylinder cover is made to fit two cylinders and it contains ten valve chambers, five for each cylinder, there being two starting air valves for each cylinder. Cast iron cylinder liners are inserted in the bronze casing which are recessed at the upper end to afford sufficient space for properly placing the valve housings. Force feed lubrication at about 30 lbs. pressure is used for all bearings, the piston or gudgeon pin is lubricated by means of the hollow connecting rod through which the

oil passes to it. The crankshaft is hollow so that the oil may be conveyed to the main bearings in this way. All bearings are made of ample dimensions, the gud-

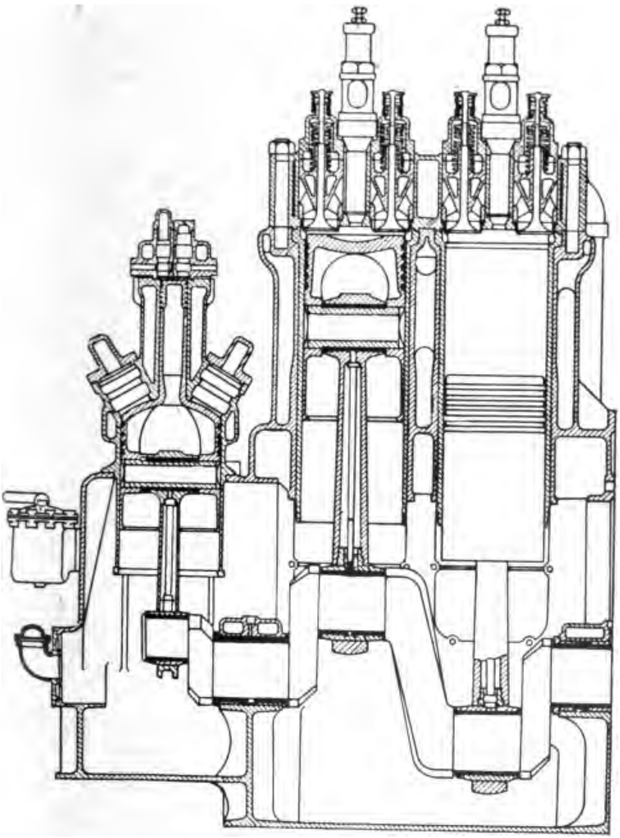


FIG. 69

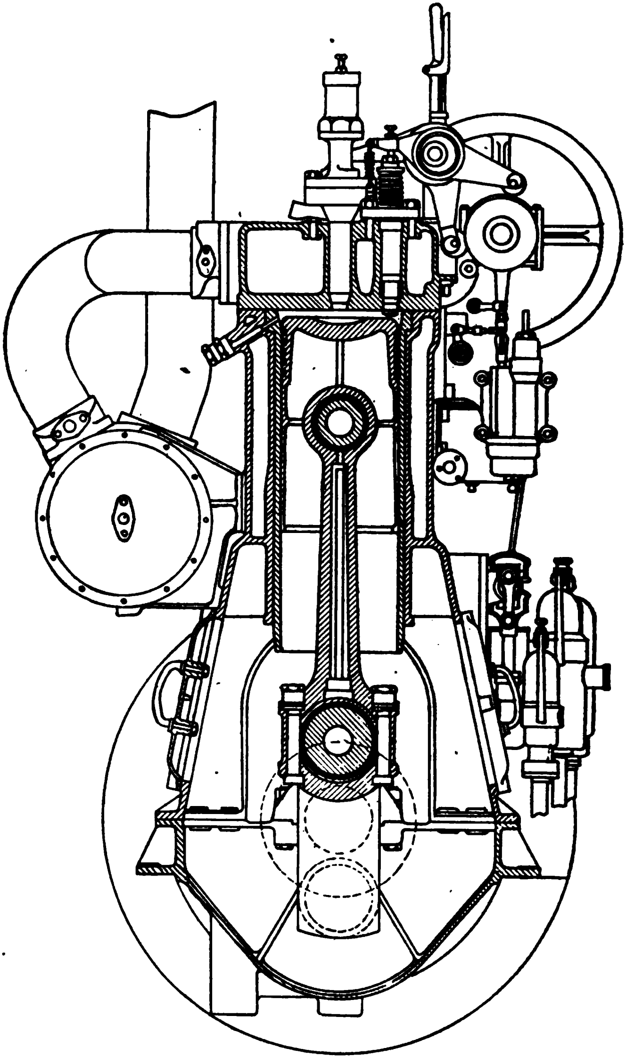


FIG. 70

geon pin being of specially large diameter. For each pair of cylinders the crankshaft has crankpins set at 180°. The two-stage single-acting air compressor is placed on the end of the crankcase and is operated by the overhung crank of the main crank shaft. Reversing the rotation of the engine is effected by the longitudinal movement of the camshaft, which in this type is equipped with two sets of cams, the one for going forward and the other for astern; the side movement of the camshaft is performed by the operation of a handwheel. The use of two starting valves in each cylinder overcomes the difficulty of providing space for extra cams which would be necessary with one starting valve. The operation of reversing is accomplished as follows:

1. Put out of action the master air inlet valve by movement of the bell crank lever controlling the same.

2. Movement of the camshaft by two turns of the handwheel which automatically closes the air inlet valve and brings a double cam under the exhaust rocker arm, thus allowing exhaust to take place at each revolution.

3. Manipulate the lever allowing compressed air to enter the cylinder.

4. Give further movement of the handwheel controlling the camshaft motion; in this way the air inlet and exhaust valves are brought into ordinary action again, as are also the fuel inlet valves; simultaneously the starting valves are placed out of action.

The complete operation of reversing is effected by the movement of one handwheel, as already explained,

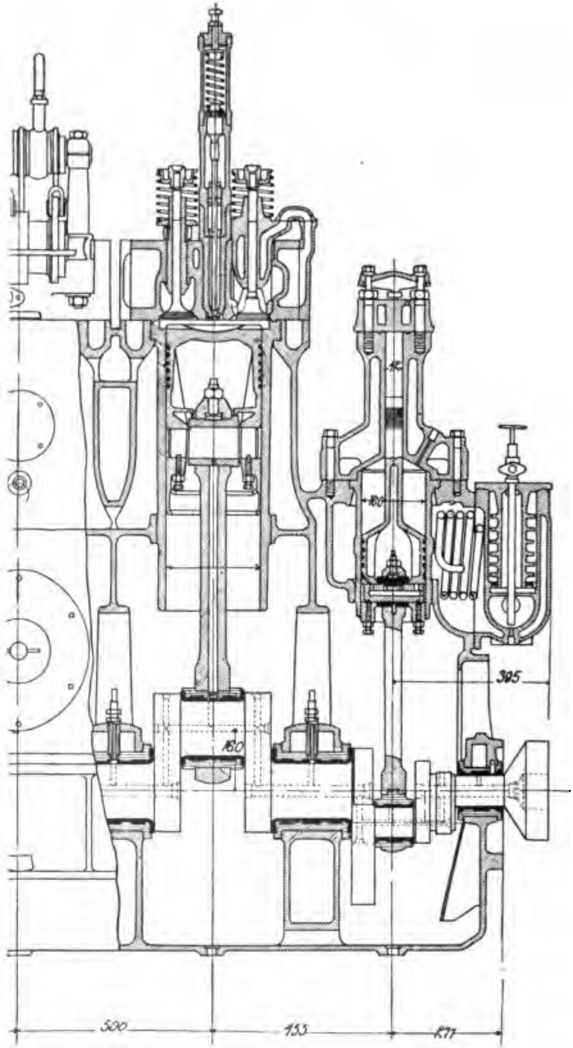


FIG. 71

together with the movement of one lever on the end of the rocker which controls both the starting valves and the fuel inlet valves. Two other levers are provided, one of which controls the air suction to the air compressor and the other controls the supply of fuel. All these levers, as well as the handwheel, are within easy reach of the attendant in the front of the engine.

DEUTZ MARINE ENGINE—Fig. 71 shows the 3 cylinder vertical Marine Deutz Diesel engine as made by the Gas-Motoren Fabrik Deutz.

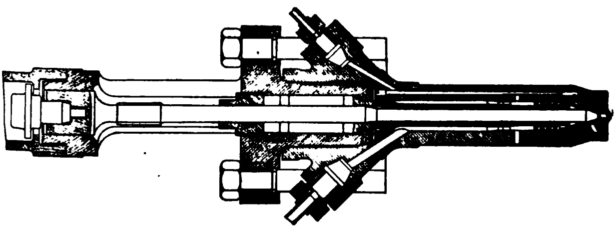


FIG. 72

It operates at 344 R.P.M., each cylinder being 240 mm. (9 7/16″) with stroke of 320 mm. (12 9/16″). It is made non-reversible and a reversible propeller is used in connection with it. The maximum power developed by this engine was 89 B.H.P., the fuel consumption then being 0.462 lbs. per B.H.P. hour; when running at 72 B.H.P. the fuel consumption was 0.471. At the maximum power the air injection pressure was 950 lbs. and was 900 lbs. at 72 H.P.

The bedplate of this type is cast in one piece and the cylinder casings and frames are also cast in one piece,

the cylinder liner being carefully fitted to each casing. A sump for waste lubricating oil is cast in the lower part of the bedplate, as shown in the illustration. The air inlet and exhaust valves each are 84 mm. (3 11/32″) in diameter and are placed in the cylinder head; they are actuated by rocker arms from the horizontal camshaft placed at the upper part of the cylinder. This camshaft is actuated by vertical intermediate shaft and gears from the crankshaft. In order to allow sufficient space for placing the valves,

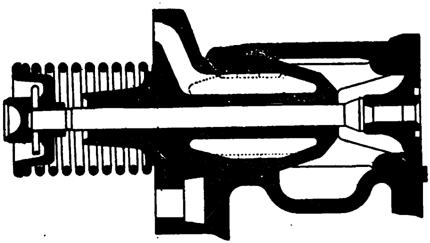


FIG. 73

the cylinder liner is recessed at the upper end. Each valve is contained in a separate housing, the exhaust valve housing being water cooled.

The spray valve, or atomizer, is shown in section at Fig. 72. It is arranged with two oil inlet passages, one being for the injection oil and the other for tar oil, or other heavy fuel. The operation of this sprayer is similar to the sprayer shown and described at Fig. 18. The two stage air compressor shown in the sectional view for furnishing high pressure injection air is

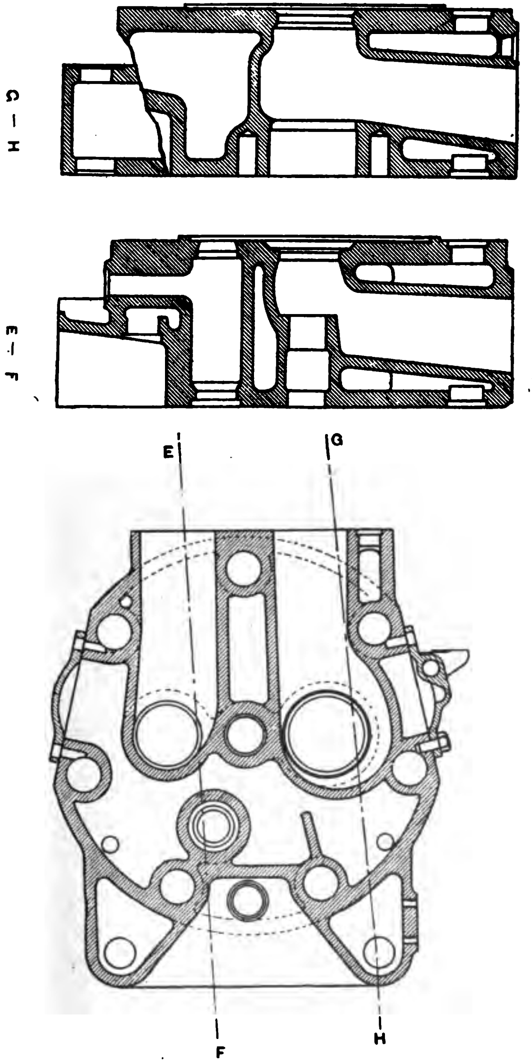


FIG. 73A

equipped with an intercooler and final cooler. All lubrication is effected by force feed pump. The crankshaft being hollow, lubricant is admitted to the main bearings, in this way, and the connecting rod is also hollow, thus allowing the lubrication through it to the piston, or gudgeon pin, to be thoroughly effective. Waste lubricant is conducted from the sump, previously referred to, to a cooler and filter, from whence it again passes to the force feed lubricator. Water circulating pump and bilge pump are both operated from one eccentric placed at the end of the crankshaft. These two pumps are situated opposite each other and are attached to the bed plate.

The water cooled exhaust housing and valve are shown in section at Fig. 73. The cylinder head is also shown in section at Fig. 73*a*.

This make of engine is a very interesting and a representative example of the smaller Diesel type marine engines. Twenty-five H.P. in each cylinder is the minimum size, which most manufacturers have found advantageous to build.

THE M. A. N. VERTICAL 2 CYCLE MARINE type Diesel engine is shown in section at Fig. 74. It was built by Messrs. Blohm & Voss of Hamburg. This engine, installed in the motor ship "Secundus," has four cylinders each of 600 mm. (23⅝") diameter and 920 mm. (36 3/16") stroke, operating at 120 r.p.m. It develops 1,350 B.H.P. This ship has twin screws, is 398'4" long, 52'6" breadth, 35 ft. in moulded depth, and with a draught of 23 ft. carries 7,750 tons. The

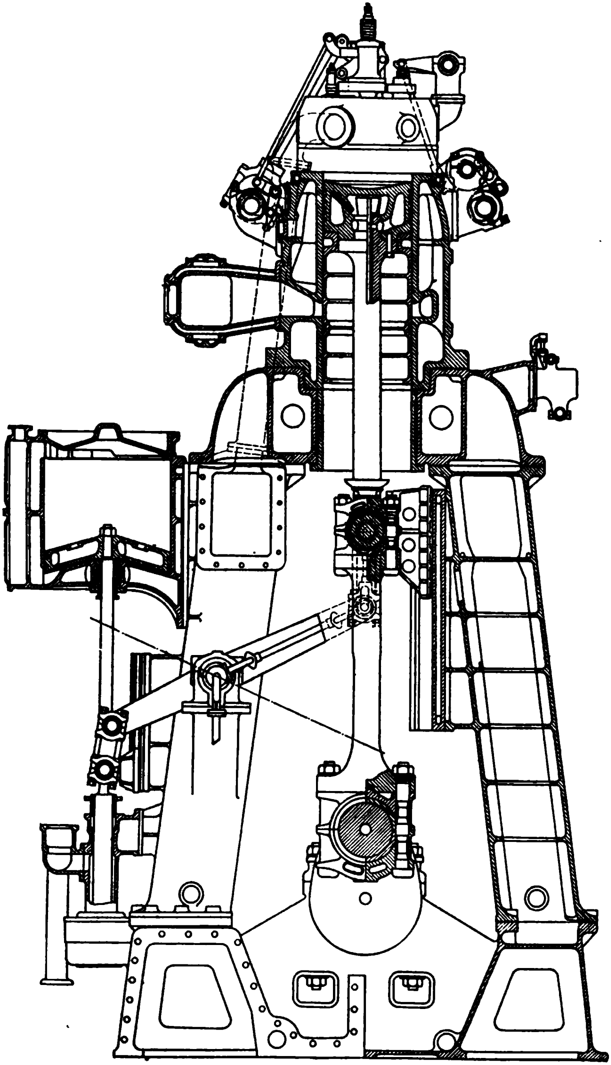


FIG. 74

fuel consumption was exceptionally low for the 2 cycle type, being 0.425 lbs. per B.H.P. hour. It will be noted in the illustration that the cylinder casings and liners are cast in one piece and are bolted to bridges spanning the vertical columns. The cylinder heads are composed of two parts, an inner water cooled shallow casting made of a mixture suitable to withstand the extreme changes of temperature and an outer casting unjacketed. This design the makers claim simplifies the casting, eliminates internal cooling stresses and is more cheaply replaced in case of fracture. The piston made in two parts is water cooled with fresh water. Force feed lubrication is furnished to all bearings. The two double acting scavenging air pumps are operated from levers attached to the crossheads. Each main cylinder has two scavenging air inlet valves actuated by eccentric rods and rolling contact levers. The camshaft is placed at the back of the cylinders. The fuel injectors are operated by cams and valve motion in the ordinary way. The injection valves open inward toward the cylinder very similar to the fuel valves on the Burmeister & Wain type, illustrated and described elsewhere. Here a separate shaft is used to operate the fuel pumps and the starting valves placed in front of the engine. The eccentric or camshaft above referred to is at the back side of the cylinders.*

The method of control is as follows—a "selector"—shaft runs the whole length of the engine which is

*A six cylinder engine of the same design is now in course of construction at the Brooklyn Navy Yard, N. Y., but the valve motion in it consists of ordinary cams and rocking levers.

moved either by hand by means of the handwheel or usually by means of an electric servo motor, which motor is controlled by a switch, the handle of which can be placed for "ahead" or "astern" running as required. This motor through worm gear and a vertical shaft causes movement of the "selector" shaft. A pointer placed behind the handwheel records on its dial the movement of the "selector" shaft so that the attendant can note the exact position of it. The two starting valves in each cylinder are actuated simultaneously by one rod attached to a built up rocker free on the "selector" shaft. Two cams, one for ahead, the other for astern, are attached to this shaft. Movement of the "selector" shaft from its neutral position brings either cam in such position that by means of arrangement of rockers and rollers the starting valves are opened on each cylinder. Movement of the selector shaft towards its neutral position allows the starting valves on successive cylinders to be put out of action. The vertical shaft actuating the horizontal selector shaft is arranged to control the suction valves of the fuel pump as well; thus when the starting valves are in operation the fuel suction valves remain open. The fuel regulator wheel placed beside the larger handwheels controls the fuel supply through a main cock on each pump box, it also controls the air blast injected with the fuel. Thus reversing is a comparatively simple operation and consists of the movement of a lever which in turn brings the valve motion to the desired position and the movement of a handwheel controlling the fuel supply. Fig. 75 shows the arrangement of

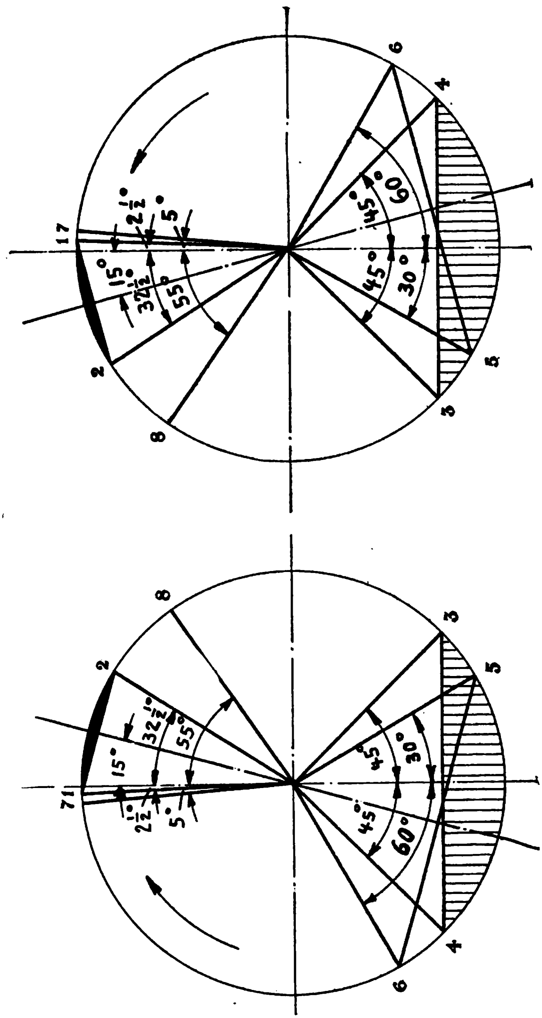
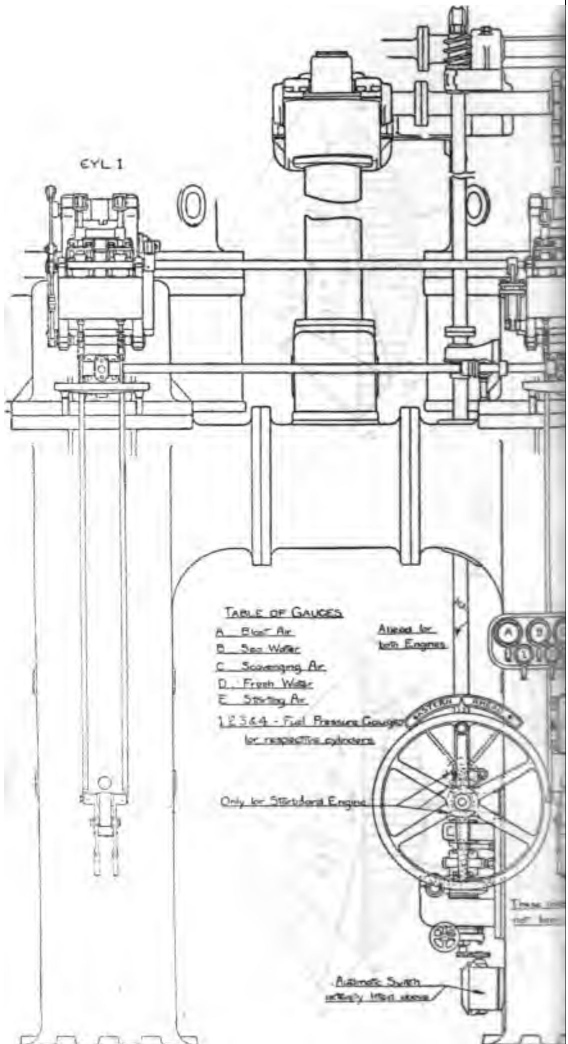


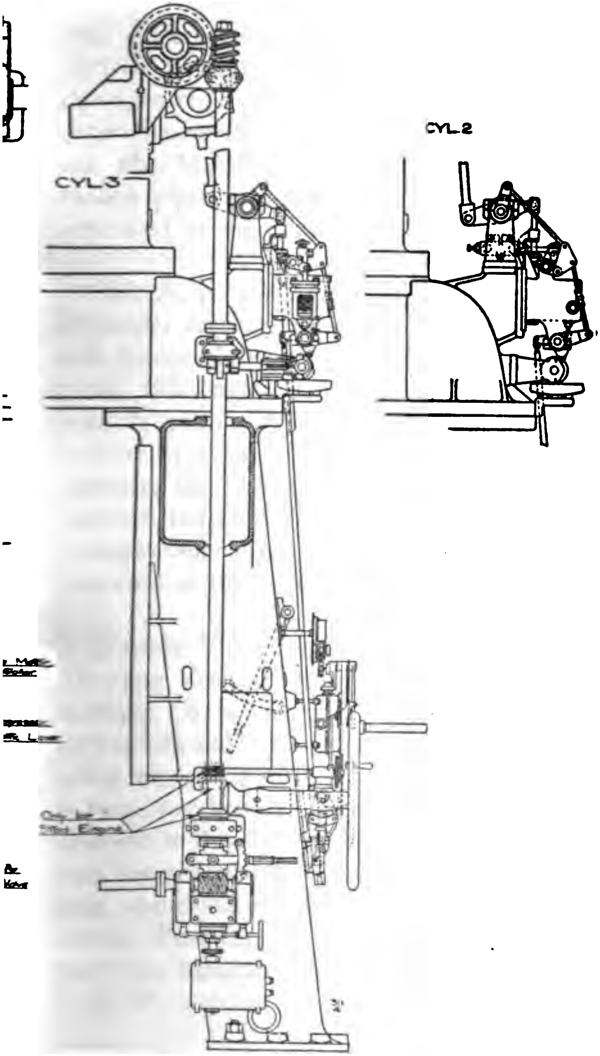
FIG. 75A

1. Fuel inlet opens.
2. Fuel inlet closes.
3. Exhaust port opens.
4. Exhaust port closes.
5. Scavenging valve opens.
6. Scavenging valve closes.
7. Starting valve opens.
8. Starting valve closes.



CYL 1
TABLE OF GAUGES
A Blast Air
B Sea Water
C Scavenging Air
D Fresh Water
E Starting Air
1 2 3 & 4 - Fuel Pressure Gauges for respective cylinders
Ahead for both Engines
ASTERN
AHEAD
Only for Starboard Engine
Automatic Switch

PLATE IV



To face page 140

the different levers and handwheels, their relative position to each other as well as the gauges and other devices required in the operation of this engine. Fig. 75*a* shows the different positions of valve openings on the M.A.N. 2 cycle marine type when reversing takes place. It is effected by the play of about 30° allowed in the clutch on the intermediate cam shaft.

M. A. N. 2 Cycle Marine.—Fig. 76 shows a six cylinder marine engine previously built by this firm of the two-cycle type. As will be seen from the illustration, this engine has an upper and lower cylinder in which pistons operate. The upper cylinder is the motor cylinder, the lower cylinder being used for furnishing the scavenging air the compressed air for injection purposes, being furnished by the two-stage air compressors placed in line with the other cylinders at the end of the engine.

Krupp Vertical Marine Engine—The two cycle Marine Vertical Diesel Engine as built by Messrs. F. Krupp Co. is shown in section at Fig. 77. It has six cylinders and is designed to develop 1,150 B.H.P. running at 140 r.p.m. and has a capacity up to 1,500 H.P. when overloaded. The design of cylinder head as shown in this illustration has now been abandoned and has been replaced with a separate cylinder head casting somewhat similar to the cylinder heads shown in other illustrations. The high pressure air is furnished by two duplicate auxiliary Diesel engines each of 275 B.H.P. and air compressors capable of delivering a

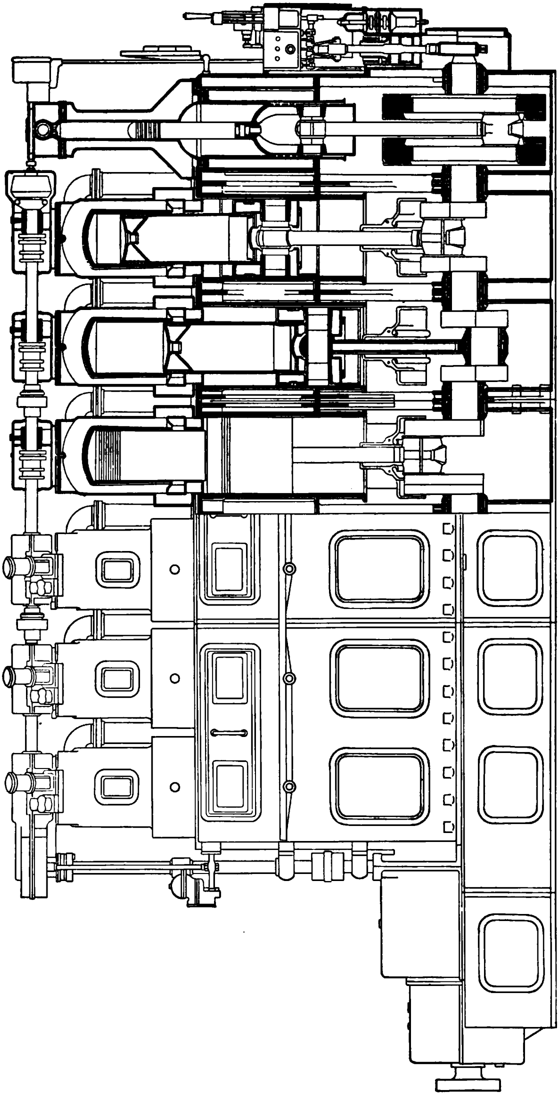


FIG. 76

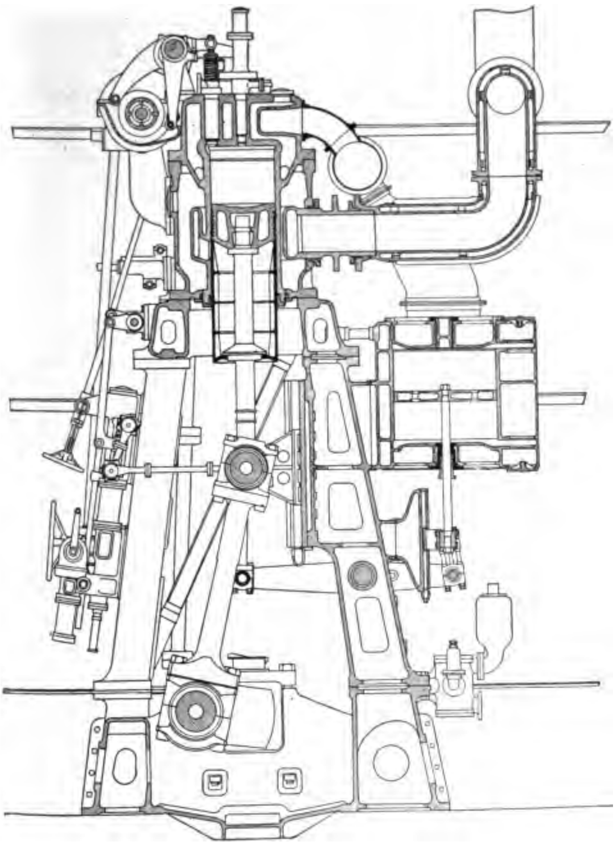


FIG. 77

pressure of 75 atmospheres. Only the scavenging air pumps and other small pumps are operated directly from the main engine.

The two double acting scavenging air pumps are placed overhead as shown in the illustration and are operated by levers from the main crosshead and are equipped with automatic valves. The bedplate is cast in three sections to which the columns supporting the cylinder plate are bolted. Each cylinder casing is a separate casting. The scavenging air is admitted through two inlet valves placed in each cylinder head; the exhaust is effected through the ports in the cylinder uncovered by the movement of the piston. The piston is made in two parts, the upper part is cooled with fresh water. The fuel is fed to each cylinder by a separate fuel pump. Each engine is arranged with three cylinders in a group and is independent; that is, they can be operaated while the other three cylinders turn idly. Reversing is effected by the movement endwise of the camshaft. Two sets of cams are provided —one for ahead and the other for astern. The reversing process is as follows:

1. Turn the handwheel to "stop" position when all valve rockers are clear of all cams;

2. Manipulation of the lever in the quadrant controlling the pneumatic plunger bringing same to "starting," thus rockers are then in contact with air inlet and starting valve cams;

3. Further movement of the lever to "running" position when all the rockers, including that controlling the fuel inlet, are each in contact with their cams, but the starting valves are placed out of action.

Fulton Manufacturing Co.: The Diesel engine

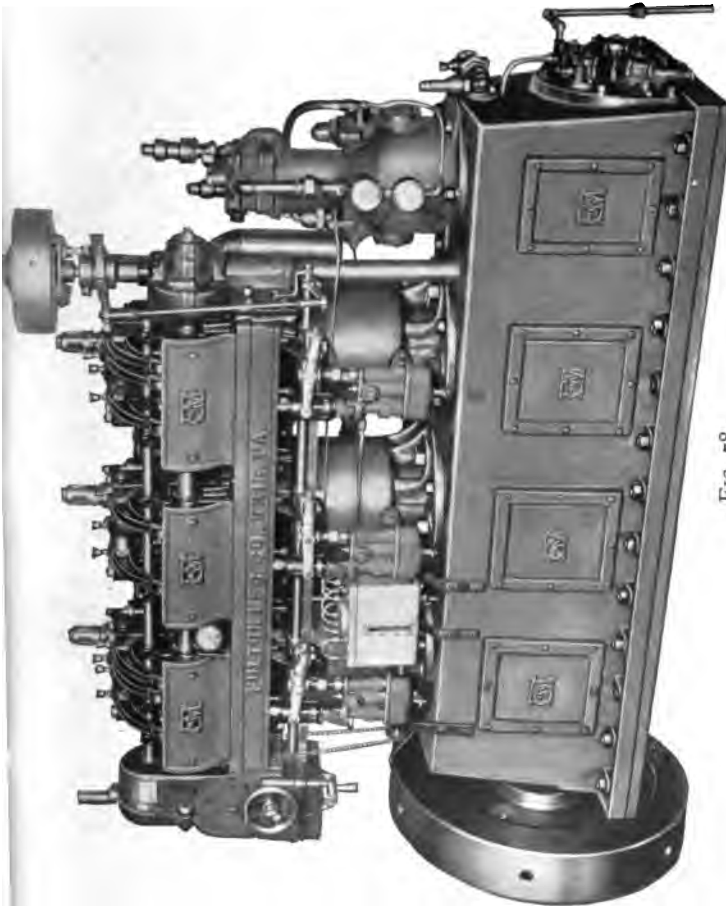


Fig. 78

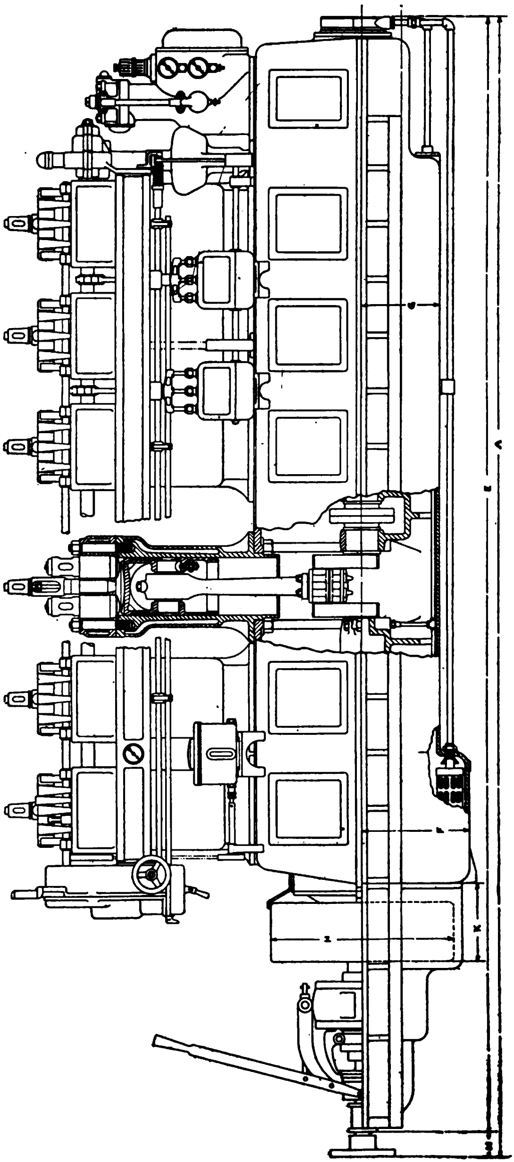


FIG. 79

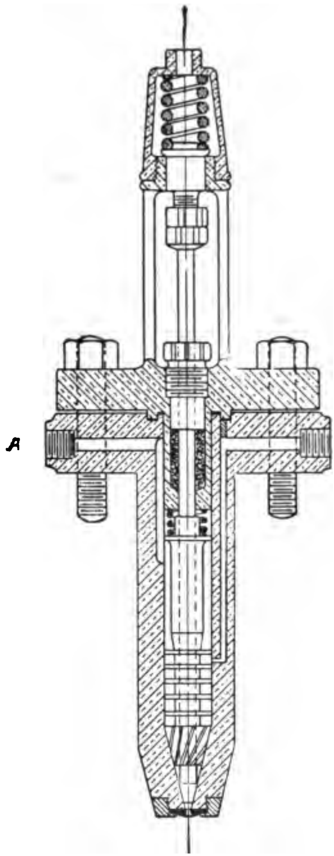


FIG. 80

as made by the Fulton Mfg. Co., Erie, Pa., is shown in the various illustrations figures 78 to 84. It works on the four cycle principle and is of the non-reversible

type, reversing being accomplished by the gear shown at Fig. 79.

Fig. 78 shows a 50 HP. three cylinder type, each single acting cylinder being 8″ diameter, 9″ stroke. It operates at 400 R.P.M. and the total weight of this engine is approximately 5000 pounds. Engines of the same dimensions are made in the four cylinder—75 HP. and six cylinder—100 HP. Representative indicator card of this engine is shown in

FIG. 81

Fig. 81 where the compression was 520 pounds per sq. in.; mean effective pressure, 90 to 100 pounds; fuel consumption being slightly under ½ pound, when using Pennsylvania fuel oil 0.864 specific gravity 18500 B.T.U. The speed of this engine is very flexible and can be reduced to 100 R.P.M. The air compressor for furnishing the high pressure compressed air is operated directly from the main crankshaft and is shown in section with the intercooler at Fig. 82. In each cylinder head are four valves, namely air inlet, exhaust, fuel, and starting. The center cylinder

has only three valves, the starting valve being omitted. The camshaft is actuated by helical worm gears from the crankshaft through a vertical intermediate shaft

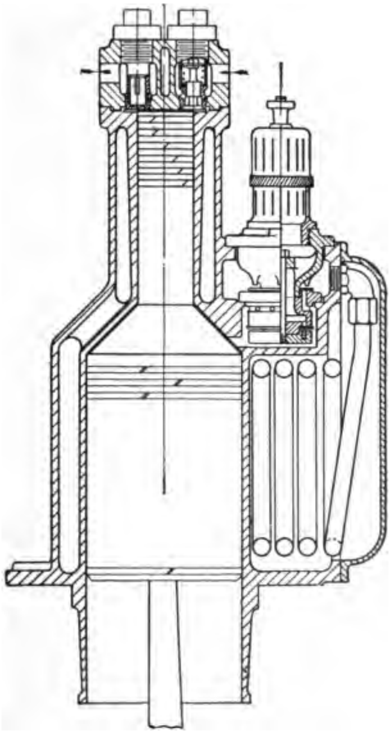


FIG. 82

gear to a horizontal camshaft placed at the top of the cylinder. The valves are operated by rocker-arms in the usual way.

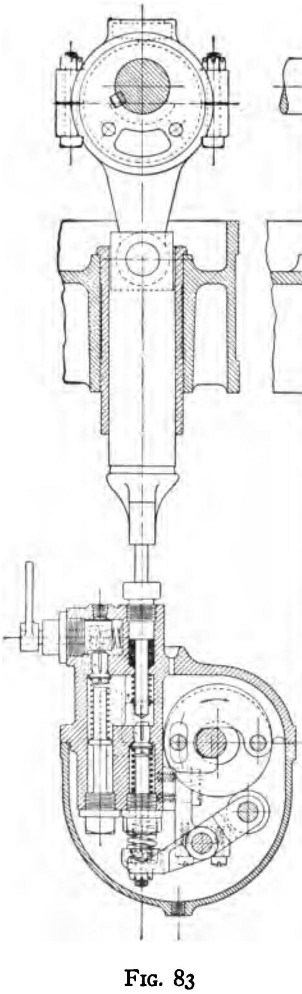


FIG. 83

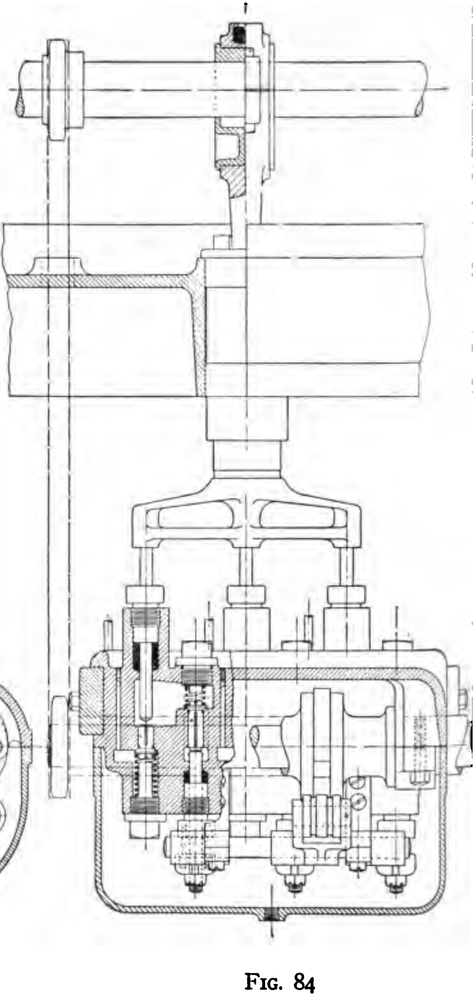


FIG. 84

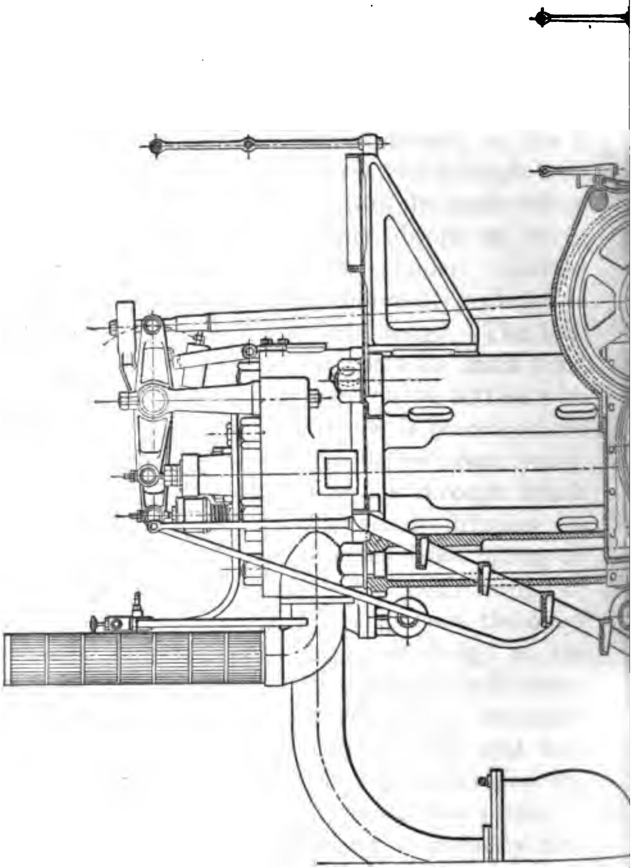
The governor which is mounted on the vertical intermediate shaft regulates the fuel supply by operating on the pump valves, varying the amount of fuel delivered by a by-pass arrangement. The oil supply pump has a separate plunger for each cylinder and is shown in Figs. 83-84. The sprayer is shown in section at Fig. 80. It is fitted with a renewable burning plate held in place by means of a cone-shaped nut automatically locked when placed in position.

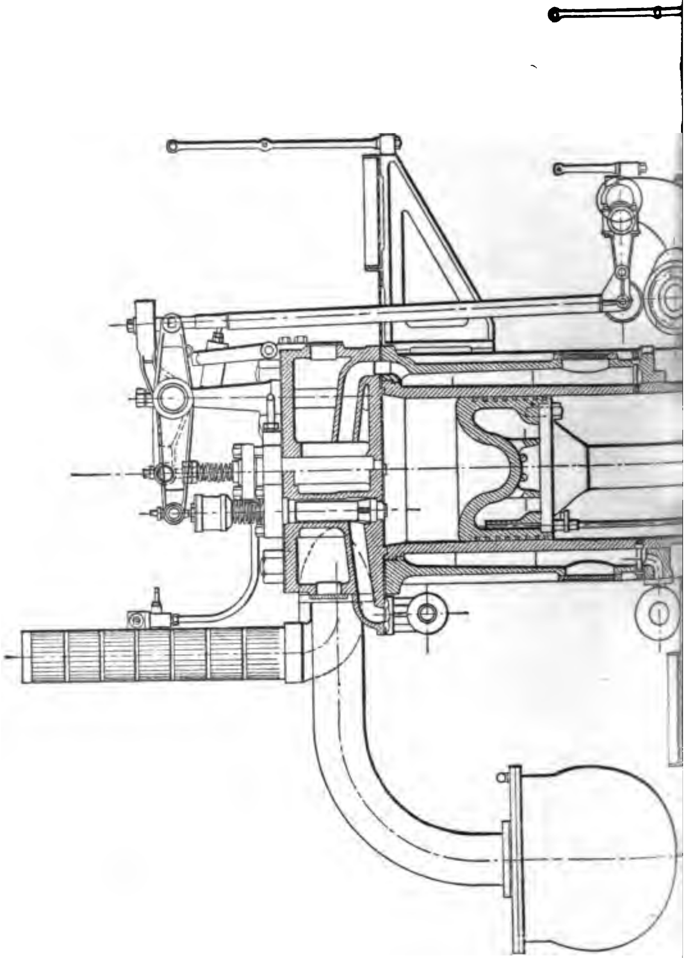
THE BURMEISTER & WAIN 4 CYCLE MARINE DIESEL ENGINE as built by that company in Copenhagen, Denmark, is shown in the illustrations Fig. 85 to Fig. 89. The engine shown is one of the six cylinder type of their latest design and installed in the motor ships "Malakka" and "Tongking." In the earlier Diesel motor ships built by this company eight cylinder engines were installed. The remarkable success obtained by these ships in the many long voyages undertaken by them have, however, led these well-known builders to install six cylinder engines instead of eight, the cylinders in the later engines being of larger dimensions and thus they develop greater power with the six cylinder unit than that developed in the eight cylinder engine. In the "Fionia," a motor ship completed just previous to the building of their latest ships above referred to, six cylinder units were installed, each cylinder having 29 9/64″ (740 mm.) diameter and stroke 43 5/16″ in(1,100 mm.) developing when running at 100 r.p.m. 2,000 indicated H.P. or approxi-

mately 1,600 B.H.P. in each six cylinder unit. The general design is shown in the various illustrations.

The A frames which support the cylinder casings have cross pieces or trestles spanning the main bearings to which the crosshead guides are bolted. Three of the cylinder casings are cast in one piece, each fitted with separate liner, and are placed directly on the A frame casting. They are furnished with through bolts passing right from the cylinder top to the underside of the main bearing. These bolts thus absorb the whole strain due to the impulse in each cylinder. Greatest rigidity of construction and maximum accessibility to moving parts is obtained with this design. The bedplate is a simple casting with supports for main bearings and side members and is closed on the bottom with a steel tray bolted in place. A cover is placed on the upper part of each A frame, below the cylinders, which is equipped with a simple stuffing box through which the piston rod passes. By this means dripping oil or water from the piston is prevented from falling on the main bearings and crankpit. A circular trough or tray is fitted to catch any oil drippings from the outer walls of the cylinder. The improved design of the piston is shown in Fig. 86. It is cooled with salt water, the circulation to and fro being effected by telescopic tubing furnished with suitable drain pipe and tray around the gland. This system has superseded the previous method of oil cooled pistons. The piston as now made is of sufficient length only to properly contain the eight piston rings.

The cylinder head containing air inlet, exhaust, fuel





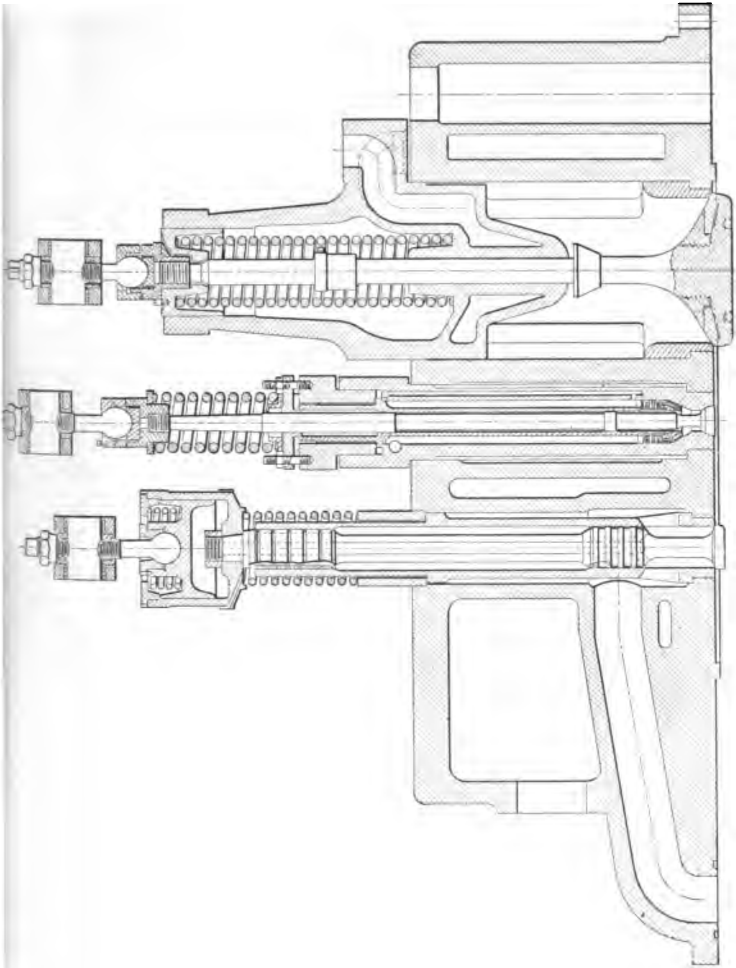


FIG. 87

injection, and air starting valves is shown in section at Fig. 87. The patented fuel injection valve opens inwards towards the piston. It has a coned head designed to uniformly spread the fuel thoroughly all around in the combustion space; in this way the heat of combustion is evenly distributed and is not felt entirely in the centre of the piston. A second coned surface on the valve forms its seat and takes the full pressure from the impulse in the cylinder, the first named coned surface being simply to spread the fuel as described and ensure thorough and proper pulverization.

The exhaust valve stem and head are made in one piece. The seating ring composed of a special mixture of cast iron is a separate piece. It is held in position by a threaded locking cap and can easily be replaced if necessary. A collar is fitted to the valve stem to prevent the valve falling into the cylinder in case of breakage. A coned shield is placed below the valve stem guide to prevent grit coming in contact with the sliding surfaces.

The air starting valve through which air at a pressure of about 300 lbs. enters the clinder when starting is operated from the starting cam and rocker by a ball jointed connection as shown in Fig. 87. It is fitted with four packing rings placed just above the air inlet passage. Its operation is as follows: When the main air valve is opened compressed air enters the chamber around the guide and through the hollow stem enters also the cylinder above, and acting on the piston attached to the top ball joint forces it upwards and brings the rocker arm in contact with the starting

cam, as the air pressure on this valve piston is sufficient to overcome the spring pressure. After the starting air is shut off the next contact with the starting cam is only sufficient to release the air pressure below the valve piston then the spring forces the piston downwards and thus lifts the rocker clear of cam and renders it inoperative.

The camshaft placed in the lower part of the cylinder is actuated by a chain of gearing as shown in Fig. 85. It is fitted with two sets of cams, one for "ahead," the other for "astern." Each cylinder is supplied with fuel from a separate pump. The six fuel pumps are divided into two sets and are placed near the control station. Each set can be regulated by lever control. Force feed lubrication is furnished to all bearings, crosshead and piston. The waste lubricating oil is cooled, filtered and used continuously. The exhaust gases are emitted from two exhaust funnels, one for each engine, thus improper combustion can more readily be detected and the cylinder not working properly located.

The reversing process and maneuvering is effected by a separate motor or engine instead of the rotary gear on previous ships. The rollers are first lifted from the cams by means of the layshaft, which is effected by worm gear drive; the camshaft is then by the same agent moved lengthwise to the position for ahead or astern as desired.

The regulator and reversing lever are so arranged with an interlocking device that they cannot operate simultaneously. The reversing lever controls a three-way air valve connected to the motor for "ahead" or

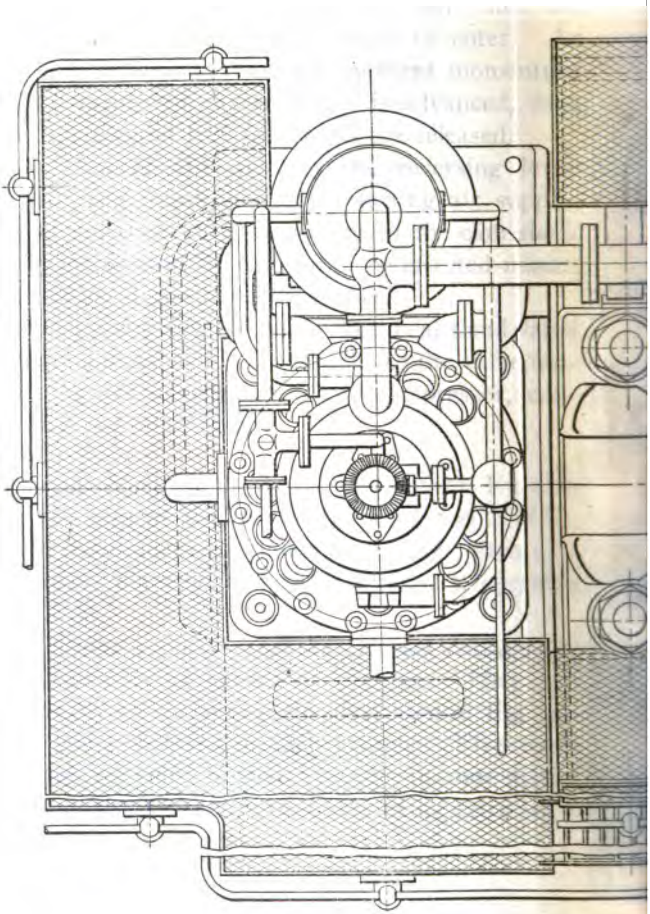
"astern." The regulator controls the admission of starting air to the main cylinders and the change over from air to fuel; it also shuts off the high pressure air to the fuel injection valve when placed in the furthest backward position as well as closing the fuel valve. No fuel or high pressure air can enter the cylinder while starting air is allowed to enter. As soon as the engine has obtained sufficient momentum from the starting air the regulator is advanced, thus the injection air and fuel feed cocks are released.

This movement also releases the reversing lever which falls back, cutting off the starting air supply. In case of emergency the movement of the cam shaft ordinarily performed by motor can be effected manually by means of the large handwheel.

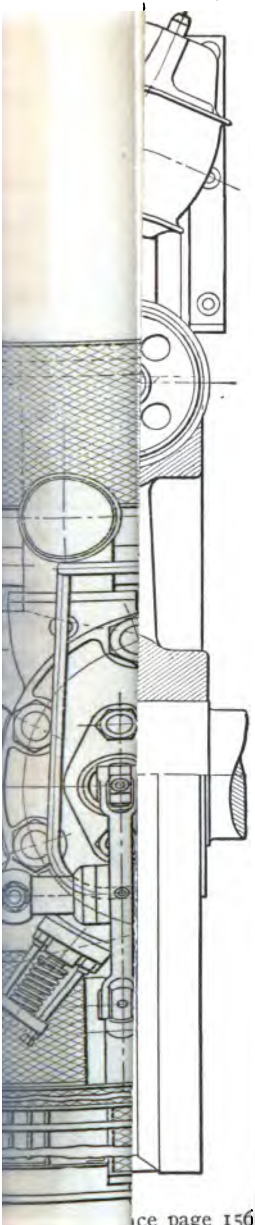
Fig. 88 shows a view of the cylinder heads and valve motion from above, while Fig. 89 shows the valve motion starting arrangements, the air compressor, connecting rod and other details.

McIntosh & Seymour Type "A" Diesel Engine —Only one size of this type is built of 500 B.H.P., having four cylinders, as shown at Fig. 90; each cylinder is 18⅞" diameter and 28⅜" stroke. It operates at 165 R.P.M.

The A-frame supports and the cylinder casings are separate castings, as shown at Fig. 91. They are bolted directly to the base plate illustrated at Fig. 92, the liner being inserted in each cylinder. Five main bearings are provided in the baseplate, the one at the flywheel end being larger to take care of its extra load.



ATE VII



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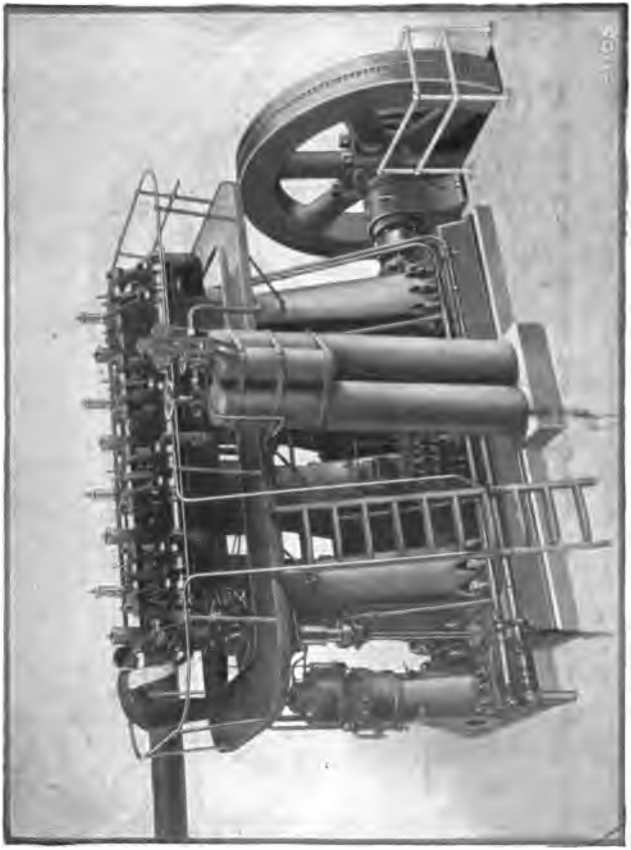

Fig. 90

They are lined with white metal and have chain lubrication. The crankshaft and connecting rod are shown at Fig. 93.

Fig. 91

The space between the A-frame and the openings at both ends are fitted with planished steel covers, thus no oil is thrown from the bearings or moving parts. In the special construction of this engine the cooling water passes from the cylinder jacket to the head

Fig. 92

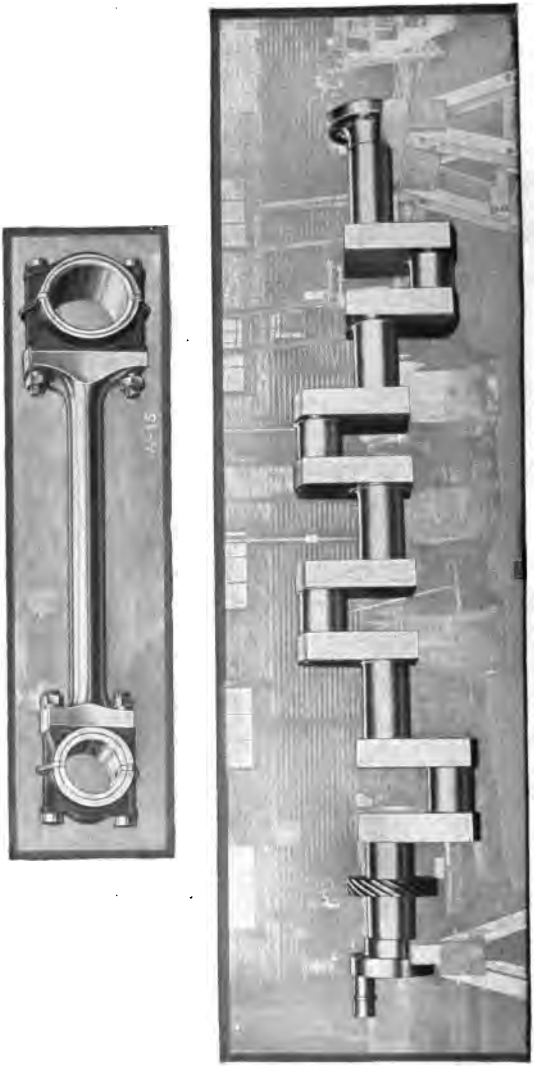

FIG. 93

through ports left in the castings of the cylinder head; a uniform cooling effect is thus obtained throughout. This design also allows easy removal of scale which sometimes forms in the water jackets. Hand holes

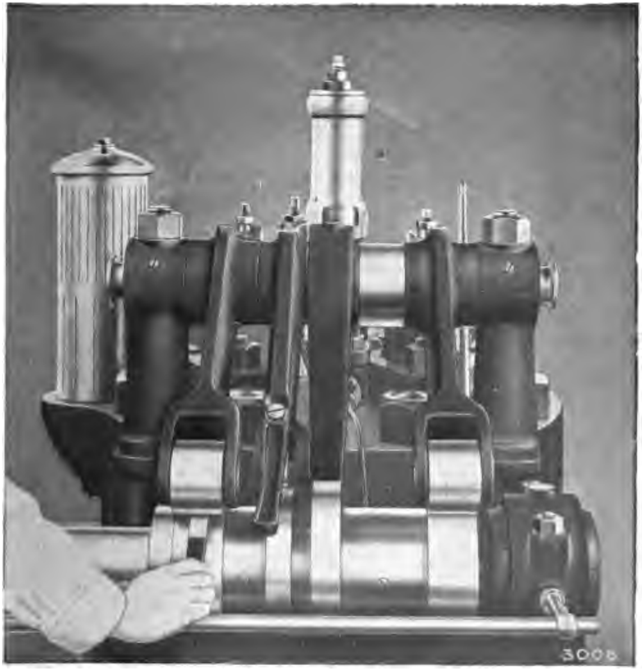

FIG. 94

are also provided near the bottom of the cylinder casing. The inner joint of the cylinder head which is exposed to the heat of combustion is a recessed joint; the outer joint and those around the studs are made

with rubber rings. An arm, shown in Fig. 91, is provided to support the pedestal for the camshaft.

In each cylinder head are three valves; namely, the air inlet valve, fuel injection valve and the exhaust valve. No relief valves are used. A starting valve is provided in the head of one cylinder only. The camshaft placed at the front of the cylinder heads is actuated by a vertical shaft taking its motion through spiral gearing from the crankshaft. Cam rocking levers, as shown at Fig. 94, are mounted on stub shafts of each cylinder head. Each stub shaft is supported on two pedestals, having caps, and has a groove at each end to facilitate removal. To take out the valves is thus a simple matter; the caps are first removed from the pedestal, the rocker gear can then be lifted without disturbing any adjustments. The piston and piston rings are shown at Fig. 95. To inspect the piston or rings it is only necessary to disconnect the fuel supply pipe, the exhaust and the two water connections, and lift the cylinder head with the rockers and shaft; no adjustments are thus disturbed. Each valve and housing can be separately withdrawn. The exhaust valve shown in Fig. 96 has a renewable seat and the guide is water cooled, which permits satisfactory lubrication.

The starting valve on this type of engine is operated by a lever which shifts the narrow roller in contact with its cam. In the "off" position the roller of the starting valve rocker clears the cam and it has no motion. The air and exhaust rollers and cams are made very wide to afford ample rolling surface. They are composed of chilled cast iron.

FIG. 95

A special injection valve is used in this design. Instead of the usual perforated discs, a pulveriser cone in which passages are drilled to form a number of sharp angles is used; thus the fuel and air are thoroughly intermingled, and when the pulverised fuel mixture en-

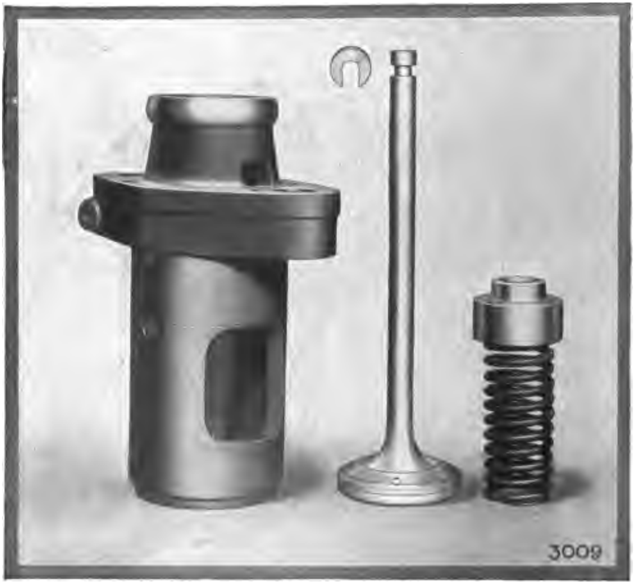

FIG. 96

ters the cylinder through the spirally drilled holes of the burner plate it forms a cloud which can be easily burned. Each cylinder has a separate fuel pump, the four pumps for the engine being fitted in pairs in two right-angled lines. Each pair is contained in two

adjacent chambers placed in the middle of the pump barrel. In this position they are very accessible and can be removed when the gate which holds down the cap has been slacked, or swung around. A single eccentric operates all the pump plungers. Those nearest the eccentric are operated directly, those at the back-end of each barrel are driven by return rods; a small cylindrical crosshead is fitted both back and front of each line of the pump. The eccentric that drives the fuel pump plungers is arranged to float upon another eccentric. The action of the governor alters the relative positions of the two eccentrics, thus the stroke of the pump is lengthened or shortened, producing variations required in the fuel fed to the cylinders. The pump and the governor are actuated from the vertical intermediate shaft.

For the supply of air for injection and starting a two stage compressor is employed. The air inter-cooler is a water jacketed chamber with ribs provided to facilitate heat transfer; any moisture which is drawn from the atmosphere or oil which may be carried from the L. P. cylinder can be blown off through a drain provided for that purpose. After passing to the H.P. cylinder and final compression, final cooling takes place. The high pressure cooler consists of a coil of copper pipe inserted in a large water jacket with its outlet at the lower end; another drain is provided so as to release any further condensation; the high pressure air is quite cool. Both the low pressure and the high pressure stages have safety valves and pressure gauges.

The main bearings are lubricated by chains dipping

into the oil baths formed below the bearings; the crank pins have centrifugal ring oilers, fed mechanically. The lubricating oil is fed to the cylinder walls at two opposite points where it fills the grooves of the pistons and is carried up and down the cylinder walls. For lubricating the piston pin, the oil is taken in close to one end through a channel drilled in the piston and another channel in the piston-pin leading to its centre; from the inner end of this channel it issues through two holes to the bearing surfaces in the piston pin bearing. The lubricating oil pumps are assembled in a group on the front of the engine and are driven from the camshaft above. The cams and rollers, camshaft, gear casing, etc., are all hand oiled.

McIntosh & Seymour Type "B" Diesel Engine—This design is built in various sizes consisting of three series of engines, constructed in multi-cylinder units of either 45 H.P., 90 H.P., or 165 H.P., in each cylinder respectively. They have closed crank cases upon which are mounted the cylinder casings. They are illustrated by the sectional and other views shown at Fig. 97, Fig. 98 and Fig. 99, which illustrates a 500 H.P. size. With the exception that the cam shaft and its bearings are completely assembled in a case and can be removed in one piece, all the details of this series of engines are practically identical with the construction of the A-frame type previously referred to in detail.

Figs. 100 to 120 are left for future additions.

FIG. 98

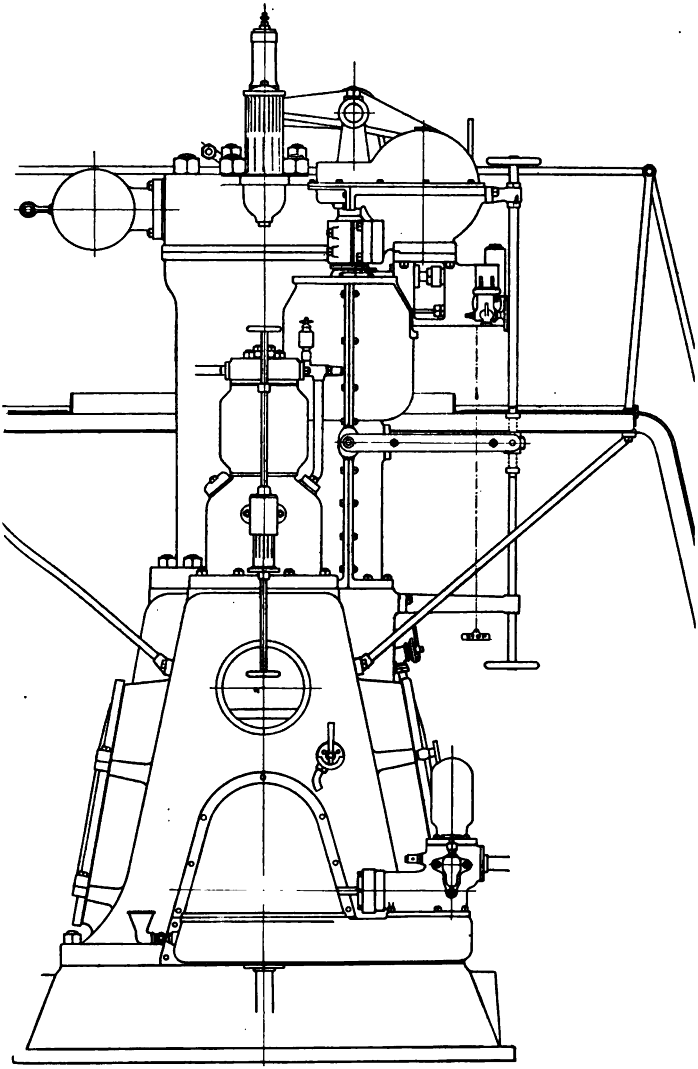

FIG. 98

TABLE III

TEST ON 500 B. H. P. McINTOSH-SEYMOUR CORPORATION DIESEL-TYPE ENGINE

STROKE 28⅜ in. CYLINDER DIAMETER 18⅞ in.

Test No.	1	2	3	4	5
Date 1915	Jan. 11	Jan. 11	Jan. 11	Jan. 11	Jan. 11
Load on brake=Lbs.	1651.5	1513.5	1150.0	750.0	370.0
Constant K	497.7	497.7	497.7	497.7	497.7
No. of R. P. M.	171	165	172	174	170
Brake H. P.	567	502	397	262	126
Load %	113.4	100.4	79.4	52.4	25.2
Time of test in hours	½	1	1	½	½
Fuel consumption during test Lbs.	115.0	204.7	163.8	58.9	41.1
Total fuel consumption per hour Lbs.	230.0	204.7	163.8	117.8	82.2
Fuel consumption per B. H. P. hour Lbs.	0.405	0.407	0.412	0.449	0 652
Injection pressure Lbs. per sq. in.	925	925	815	785	775
Exhaust gas appearance	Clear	Clear	Clear	Clear	Clear
Inlet temperature of cooling water F	56	56°	56°	56°	56°
Outlet temperature of cooling water F	145	147°	145°	147°	150°
Temperature in testing room F	62	62°	62°	62°	62°

Analysis of fuel is given in table IV.

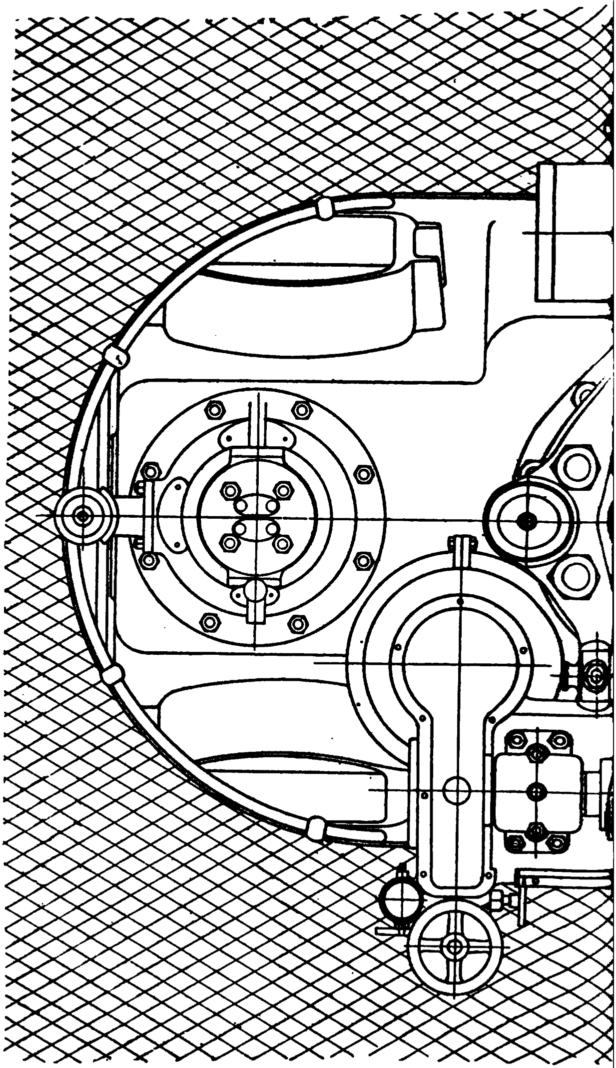

PLATE X

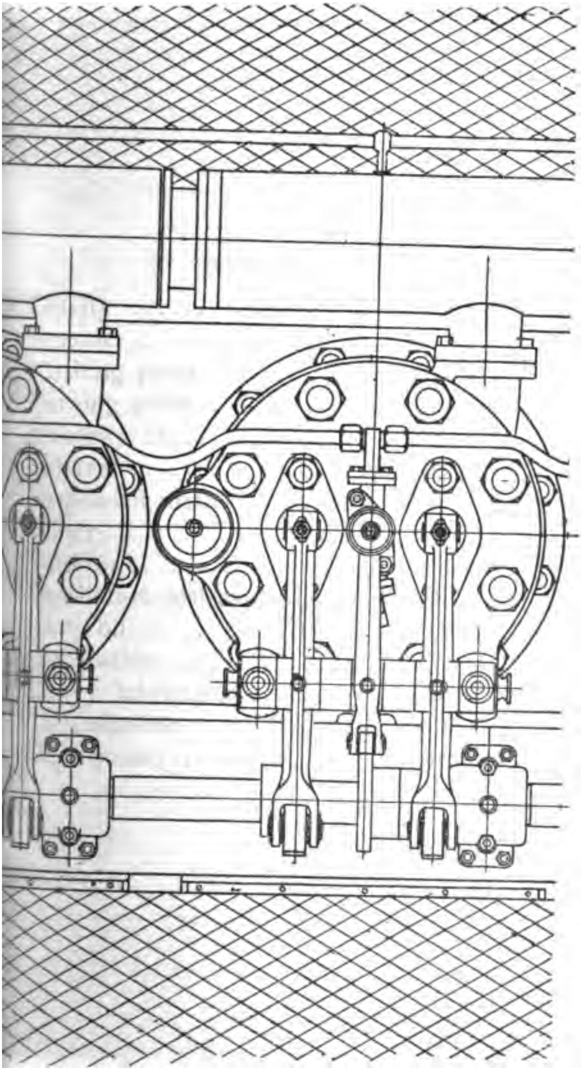

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TABLE IV

Analysis of Fuel Oil

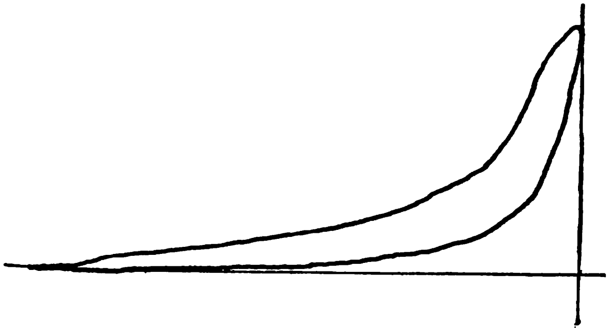
Gravity	60° F. .8550%
Beaume	60° F. 33°
Flashing point	190° F.
Burning point	246° F.
Viscosity at	68° F. 1.677%
Tar test	Negative
Suspended matter	.005%
Water	.051%
Sulphur	.187%
Distillation test-naptha	1%
Lamp oil	50.1%
Lubrication	6%
Heavy lubrication	42.5%
Carbonization	6.4%
Heat value	19266 B. T. U. per lb.

CHAPTER VIII

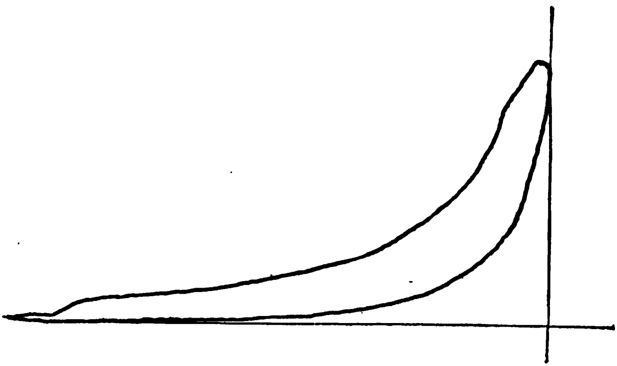
VARIOUS TYPES OF ENGINES—Continued

THE two cycle Diesel engine built by the firm of Carels Freres Ghent is shown at Fig. 122, and in section at Fig. 123 and Fig. 124. This engine, as shown in the illustrations has six cylinders 20.08 inch diameter and 30.22 inch stroke, at 130 R.P.M. it develops 1600 actual or brake horsepower. The cylinder is cast separately and bolted to the supporting A frame, a separate cylinder liner being inserted. The cylinder head cast in one piece is water jacketed, each equipped with four scavenging valves, fuel inlet valve, starting air inlet valve and safety valve. Compressed air for injection purposes is furnished by 3-stage air compressor of the Reavell type operated from the crankshaft direct (in some of the later engines the injection air compressor is operated by levers thus decreasing the overall dimension lengthwise). The scavenging air pumps are operated by levers as shown in Fig. 123. Cylinder head, and cylinder are water cooled, the piston is also cooled by oil or water circulation. Indicator cards taken from this type engine are shown at Fig. 121.

The starting or manœuvering of the engines is effected by means of compressed air furnished from air receivers, in which the pressure is maintained usually by an auxiliary engine and air compressor.



Spring 1" = 361 lbs. (1 mm. = Kg), M.E.P. = 76.1 lbs. (5.34 atm:). Injection air pressure 855 lbs. (60 atm:). Rpm. 187.



Spring 1" = 361 lbs. (1 mm. = 1 kg). M.E.P. 92.5 lbs. (6.5 atm:). Injection air pressure 995 lbs. (70 atm:). Rpm. 187.

FIG. 121

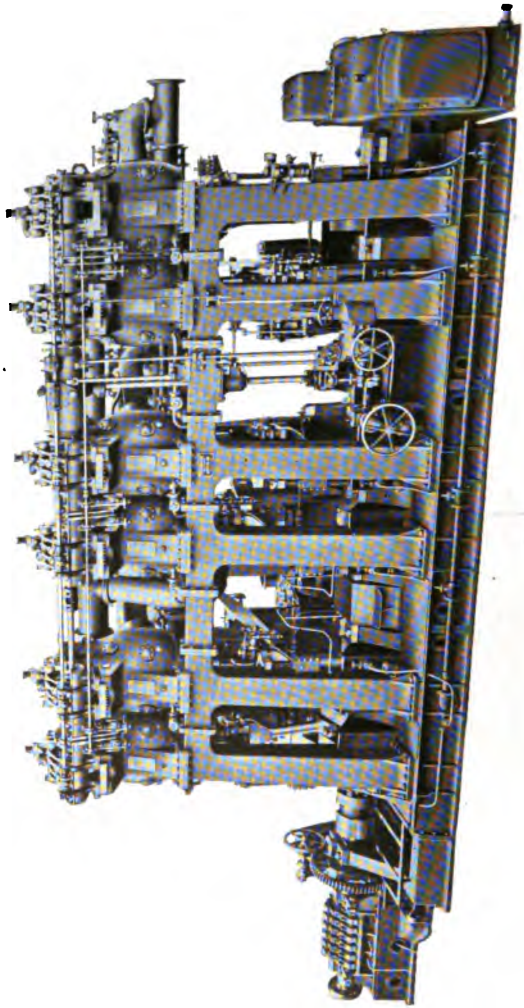


FIG. 122

STARTING—REVERSING.—The two cycle marine Diesel engine here illustrated is controlled by means of the hand wheels shown at Fig. 125. A view of a

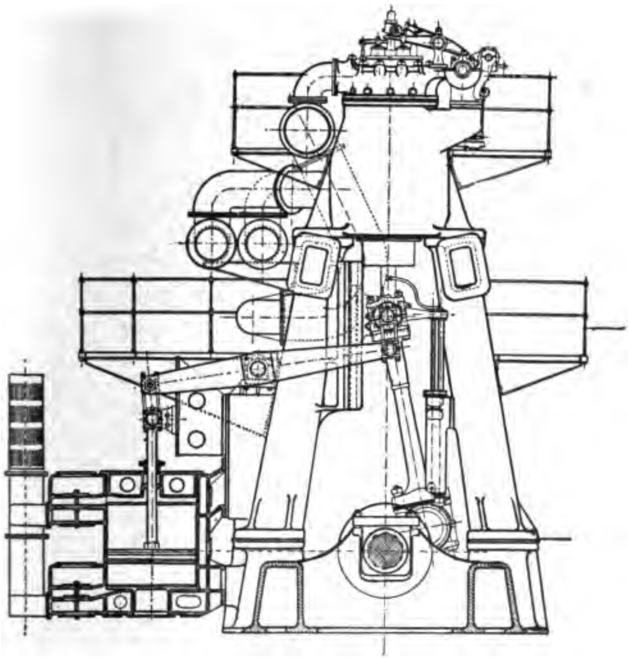


FIG. 123

part of the camshaft, manœuvering shaft, cams and valve motion is shown at Fig. 125a.

To start the engine it is necessary to raise the compressed air in the receivers to about 600 lbs. pressure which is done by means of auxiliary engine and

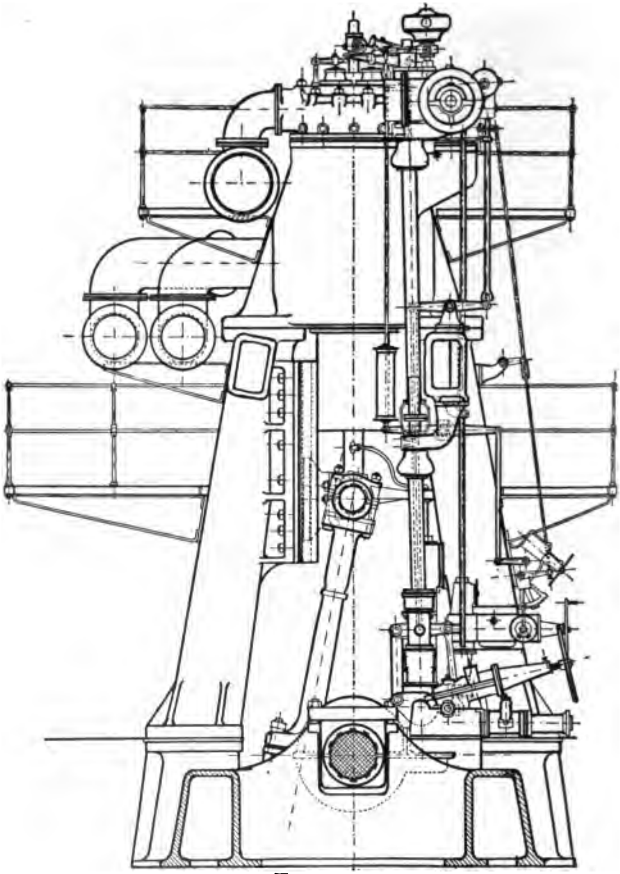


FIG. 124

compressor. The crankpin being set just past dead centre, compressed air enters the combustion space of all six cylinders. Then two (or three) cylinders have

fuel as soon as speed is attained, the remaining cylinders operating on air only. Afterwards fuel enters all cylinders, and they also come into operation in the regular way.

In Fig. 125 are shown three hand wheels or levers. That shown at 1 controls the horizontal manœuvering

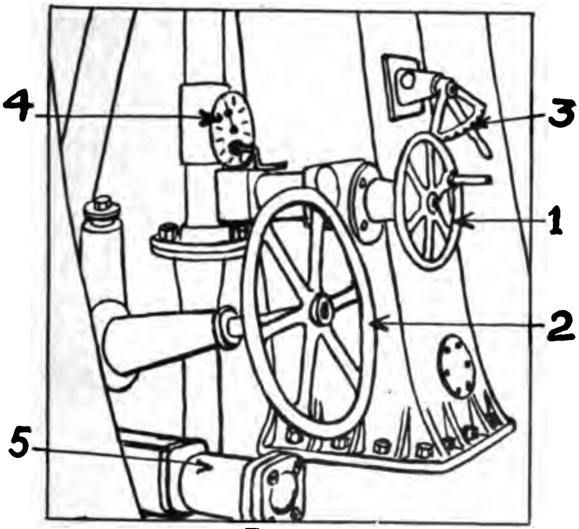


FIG. 125

shaft H (Fig. 125a) placed above the cylinders which is operated by means of a vertical shaft. The lever shown at 3 controls the air motor indicated by 5. The hand wheel indicated by 2 is provided to effect the same result and is used by hand in emergency. A dial showing what is occurring in each set of cylinders is at 4. In Fig. 125a is shown the camshaft A to which is attached

the cam C actuating the scavenger valve lever B. The lever at D controls the fuel valve and E the air starting valve. These two valves being actuated through short levers F and G and not directly from the cams. The manœuvering shaft H has a longitudinal movement and thus allows D and G, when reversing, to be brought in contact with the astern cams. Re-

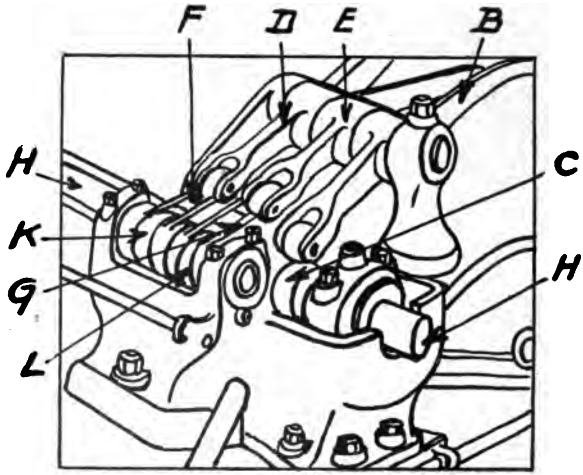


FIG. 125A

versing the direction of rotation of the crank-shaft is effected in about six seconds by turning handle 1 until indicator dial 4 points to "stop." This has turned shaft H through an angle allowing cam K to force out a small sliding part lifting the roller of the fuel valve lever. Lever 3 is now moved to operate reversing motor (this can also be

effected by hand, using wheel) which revolves the camshaft through the necessary angle to properly operate the scavenge valves after reversal and also moves the manœuvering shaft H so as to allow levers F and G to be in contact with the astern cams controlling, starting and fuel inlet valves. Next handwheel 1 is moved till dial 4 shows all six cylinders starting up on air. This is effected by still further turning shaft H, thus cam L allows sliding piece to be in such position as to hold the starting valve lever out of contact with its cam. The next movement of handwheel 1 allows fuel to enter three cylinders by still further movement of shaft H which rotates cam K, its nose then no longer forces out the sliding piece which is brought back by spring and allows the fuel valve cam through the short lever to come into contact with its roller. The starting valve for its cylinder is similarly put out of operation.

Further movement of handwheel 1 brings all six cylinders in regular operation with fuel.

With the four cycle type a complete duplicate set of cams is provided.

The process of reversing is similar in principle to that outlined above, that is, it is effected by means of the horizontal sliding movement of the camshaft and servo motors which having disengaged the cams and the valve lever rollers allows the sliding motion of the camshaft so as to bring into action the second set of cams so arranged as to open the air and exhaust valves at the proper period for reversal as well as the cam governing the oil inlet. Two or more cylinders being

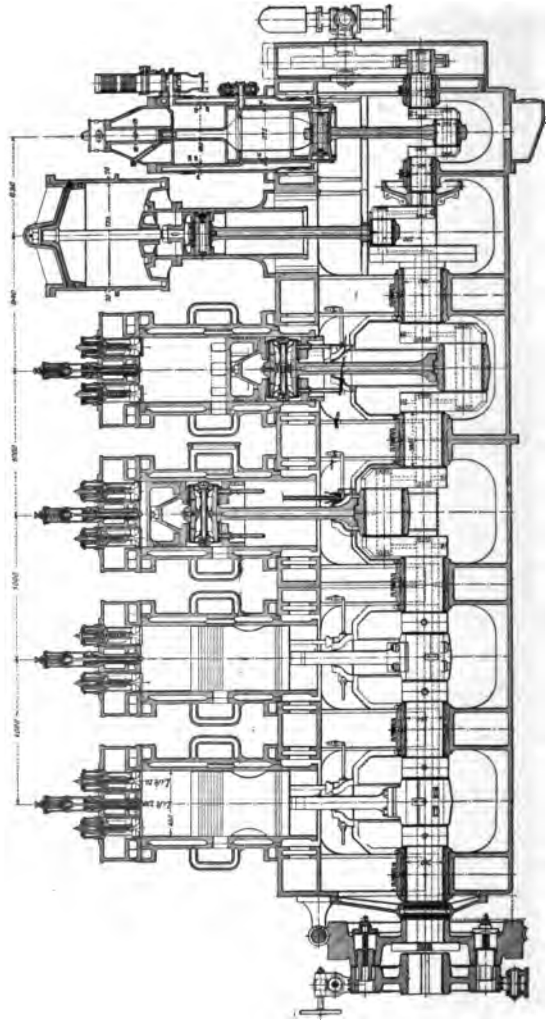


Fig. 126

operated by compressed air, while the remaining ones have fuel inlet and commence regular operation.

The auxiliary propelling engines in the cargo ship "France" are shown in longitudial section at Fig. 126, and in section through the cylinder at Fig. 127. They are of the Schneider-Carels-Diesel oil engine type of 900 actual H. P. four cylinder two cycle. Each cylinder is 17.716 inch diameter and 22.047 inch stroke and operates at 234 R. P. M. Each engine is equipped with an air compressor for fuel injection, scavenging air pump as well as cooling water pumps and lubricating oil pumps. As shown in the illustration the cylinder liners are inserted into the cylinder casings bolted to cast iron closed-in frames having large inspection doors. Guards are provided inside the frame to prevent the lubricating oil from entering the cylinder. The cylinder head is similar to that previously described, being fitted with four scavenging valves, fuel inlet valve and safety valve. The cast iron pistons are made in two parts, the cooling water or oil for same circulating through the hollow connecting bolts. As will be seen from the illustration, besides the six piston rings at the top of the piston there are two at the lower part also, to prevent escape of gases into the crank case. The valve motion is similar to that previously described for this type of engine, the fuel and starting valve, however, in this engine being operated by the same lever. Reversing is effected by longitudinal movement of the cam shaft. The three-stage air compressor for fuel injection is driven directly from the crank shaft, which also furnishes the necessary air for charg-

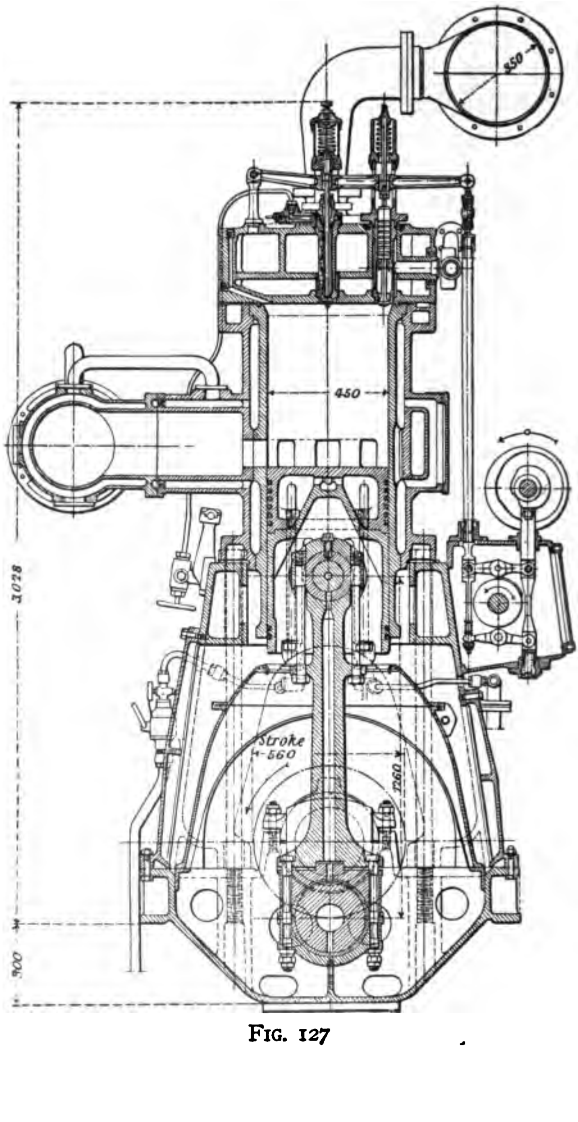


FIG. 127

ing the air receivers for starting. The piston is lubricated by a pump driven from the indicator shaft delivering the oil at two opposite points of the piston surface. The cooling medium of the pistons is circulated by a pump through telescopic tubes. A salt water circulating pump delivers the cooling water first to the air cooler, then to the lubricating oil cooler and afterwards cools the circulating oil for cooling the piston; it then circulates around the fuel injection air compressor and cylinder head. The exhaust gases pass through a water-jacketed pipe to the silencer, which is fitted with baffle plates, and from thence to the atmosphere. Air receivers of approximately 115 cubic feet capacity are charged from an auxiliary engine and compressor. The total weight of the engine is approximately 160 lbs. per actual H. P. The engine develops 1305 I. H. P. Fuel consumption 0.462 lbs., lubricating oil consumption 0.012 lbs., per B. H. P. hour.

New London Ship and Engine Company.—The four cycle marine Diesel engine as built by the New London Ship and Engine Company is shown at Fig. 128 and also in section at Fig. 129, which illustrates the specially designed valve motion consisting of two camshafts placed in bearings attached to either side of the enclosed crankcase. The camshaft on one side through a lever operates the exhaust valve, that on the other side the air inlet valve, the fuel inlet and fuel supply pump. This engine is built with four cylinders (120 B. H. P.) and six cylinders (180 B. H. P.) each being 9″ diameter and 12½″ stroke. Each engine

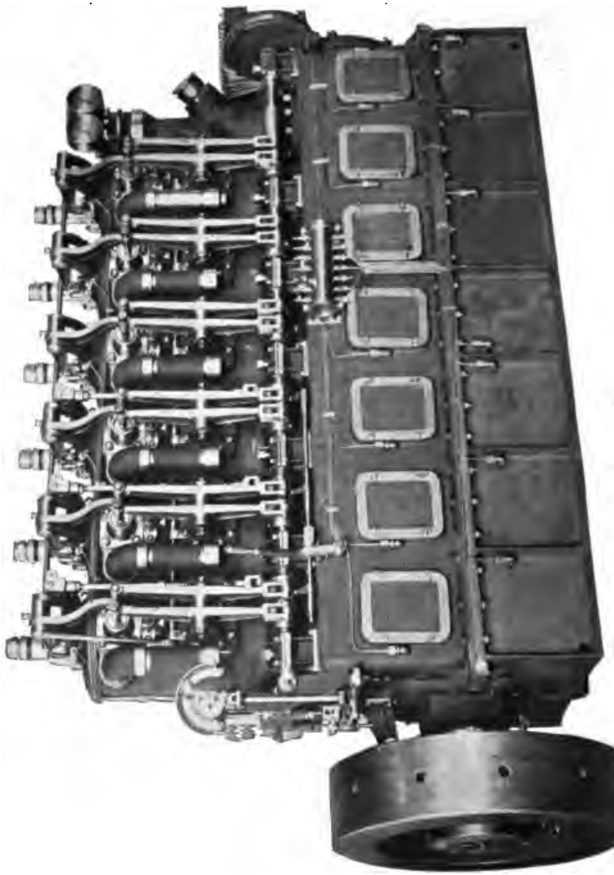


Fig. 128.

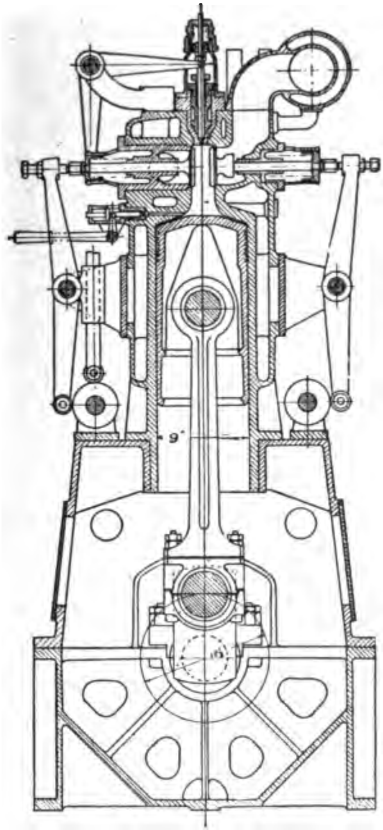


FIG. 129

operates at 350 R. P. M. The weight of the flywheel is about 2000 lbs. The total weight of the engine 8000 lbs. The engine being non-reversible, a special design of reverse gear is used. The compressed air at about 1000 lbs. pressure necessary for injection with the fuel is furnished by a 2-stage compressor placed at the forward end of engine and actuated directly from the crankshaft. The cylinders and cylinder head are cast in one piece, the air inlet valve and housing and the exhaust valve being arranged horizontally and the fuel inlet or spray valve being vertical as shown at Fig. 129. The governor placed in the flywheel acts through levers on the suction valve of the fuel supply pump regulating the amount of fuel as required by the load. The cooling water is supplied by centrifugal pump operated from the flywheel. Lubrication to all parts is effected by force feed pump.

These makers also build two cycle type marine Diesel engines with enclosed crankcase in sizes from 300 to 2000 H. P. as well as the same type with open A-frame crosshead and crosshead guides from 500 to 2500 H. P., each of these types is single acting. The latter operates at a compartively slow speed. In both types the exhausting of the gases is effected by the usual method of exhaust ports in the cylinder walls and scavenging valves placed in the cylinder head through which the low pressure air enters, thus thoroughly ejecting the burnt gases. The fuel injection high pressure air is furnished by a two stage compressor operated directly from the crankshaft, this compressor has greater capacity than is required for injection pur-

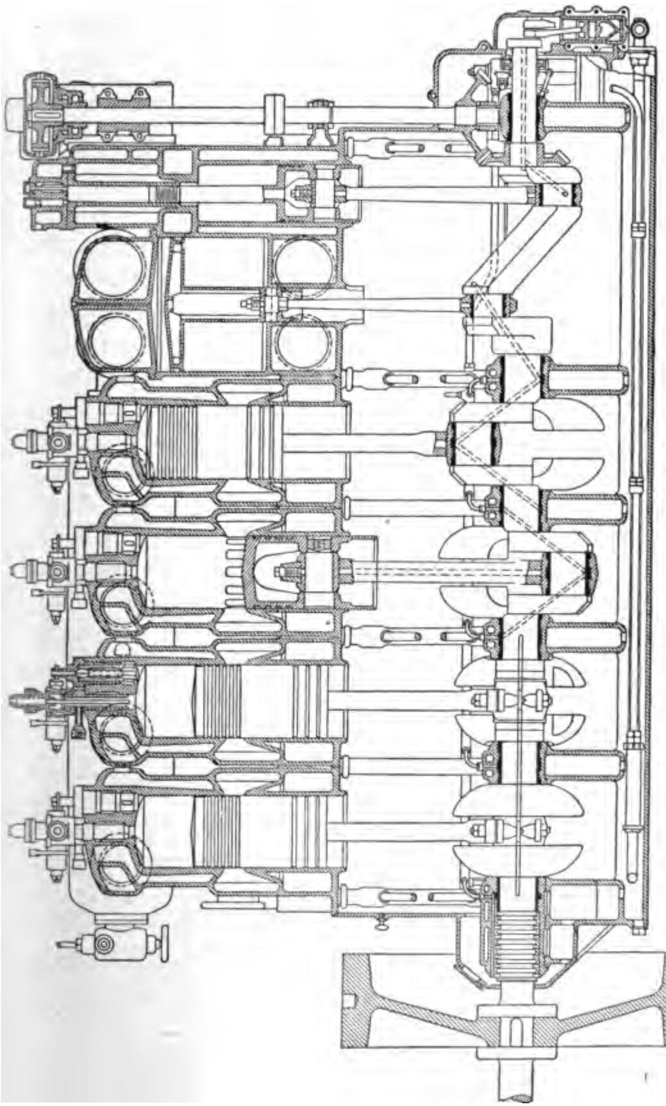


FIG. 130.

poses, the excess air being stored and is employed for starting and reversing purposes. Force feed lubrication is used throughout, the lubricant being cooled and

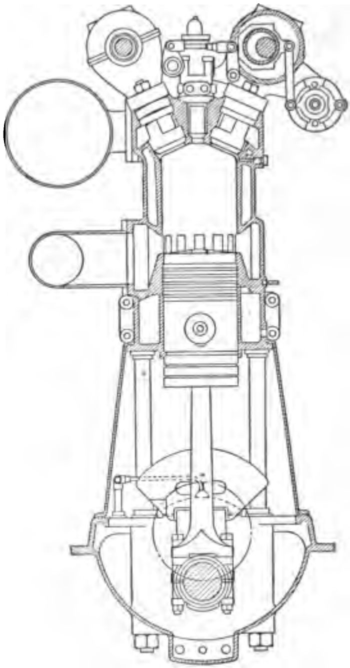


FIG. 131

contained in a closed circuit. Reversing and change of speed are controlled by one hand wheel.

The two cycle enclosed crankcase engines operate at a speed of 480 R. P. M. with the 300 H. P. and 270

R. P. M. with the 2000 H. P. six cylinder construction. Total weight is about 50 lbs. per B. H. P. The heavier type of 2 cycle engines with A frame construction operate at slower speed and weigh approximately 100 lbs. per B. H. P.

The two-cycle Diesel oil engine as made by Messrs. Sulzer, of Winterthur, Switzerland, is shown in section at Figs. 130 and 131. It has been made of the four- and six-cylinder construction. As will be seen from the illustration, it is of the single-acting type, the exhaust ports in the cylinder being uncovered by the piston at the end of its downward stroke, the scavenging air entering through the two valves placed in the cylinder head. These makers are also constructing their engines with air inlet ports, thus eliminating the scavenging air inlet valves. This engine is equipped with a double-acting air scavenging pump operated from the crank shaft and also two-stage air compressor furnishing high-pressure compressed air for injection purposes. Forced feed lubrication is provided with all bearings.

CHAPTER IX

STATIONARY ENGINES

In recent years many engineering firms in the United States have taken up the manufacture of oil engines, nearly all of them being of the Diesel cycle of operation. Some of these engines are being made of the vertical and others of the horizontal type, the former being largely made of the open crank case construction; that is, with the cylinders supported on A frames, thus allowing free access to all bearings and affording opportunity for inspection while the engine is in operation. The latter are being made by different makers both of the single-acting and double-acting type operating on the two-cycle principle and also on the four-cycle plan.

The Snow Crude Oil Engine.—The Snow Steam Pump Co. are now building two- and four-cycle horizontal single-acting oil engines operating on the Diesel cycle, and are shown in Figs. 132 and 133. The four-cycle engine shown at Fig. 132 has the air inlet exhaust and fuel inlet valves placed horizontally in the cylinder head, which are operated by cams placed on a horizontal cam shaft mounted at the rear of the cylinder head and actuated by gears from an intermediate shaft placed by the side of the cylinder. The fuel injection high pressure air is furnished by a two-stage air com-

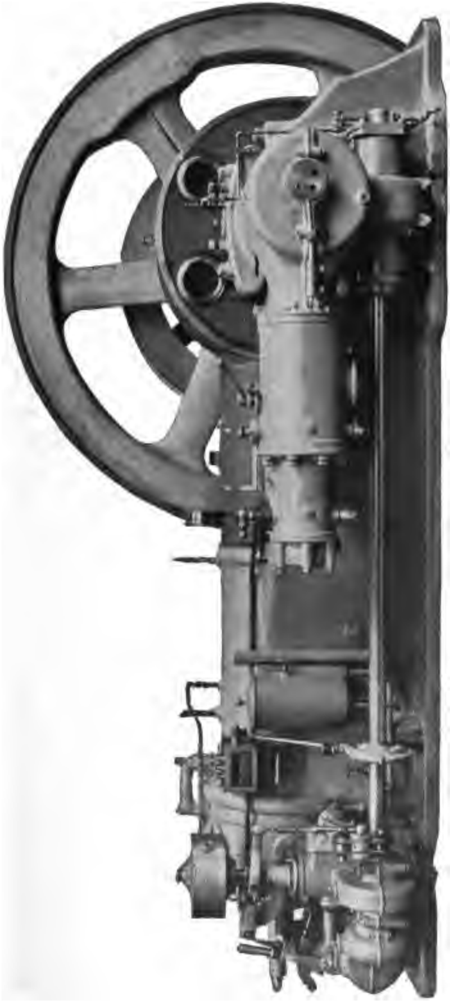


Fig. 132

pressor actuated directly from the crank shaft by crank disc. The Jahns type of governor controls the speed of the engine by operating through levers on the fuel supply pumps, lengthening or shortening the stroke of same by a wedge arrangement. The governor is mounted on the side of the main frame, and this allows easy removal of cylinder head when required. Lubrication of the piston is furnished by a Richardson positive force feed pump, which also supplies lubricant for the valve stems and air compressors. This make of engine is equipped with cross-head operating in guides placed on the main frame of the engine, the piston being shorter than the ordinary trunk type of piston used where cross-head is not employed. Reference has previously been made to the advantages obtained by the use of the cross-head.

The two-cycle type of engine is shown at Fig. 133. This engine operates on the two-cycle principle, as previously described. Exhaust ports are placed in the cylinder wall and are uncovered by the movement of the piston at the end of its stroke. In the two-cycle type scavenging air inlet valves are placed in the cylinder head with the fuel oil inlet valve; the low-pressure air necessary for scavenging is furnished by the air compressor placed ahead of the two-stage air injection compressor as furnished with the four-cycle type. The low-pressure scavenging air passes through a receiver placed in the main frame of the engine. The valves are operated by the same method as that described with the four-cycle engine and the governor operates on the fuel supply pumps in a similar way.

Fig. 133

The makers guarantee the successful operation of this engine on the lowest grade of fuel or crude oils, the fuel consumption being at full load 0.5 of a lb.; ¾ load, 0.55 lb.; ½ load, 0.6 lb.

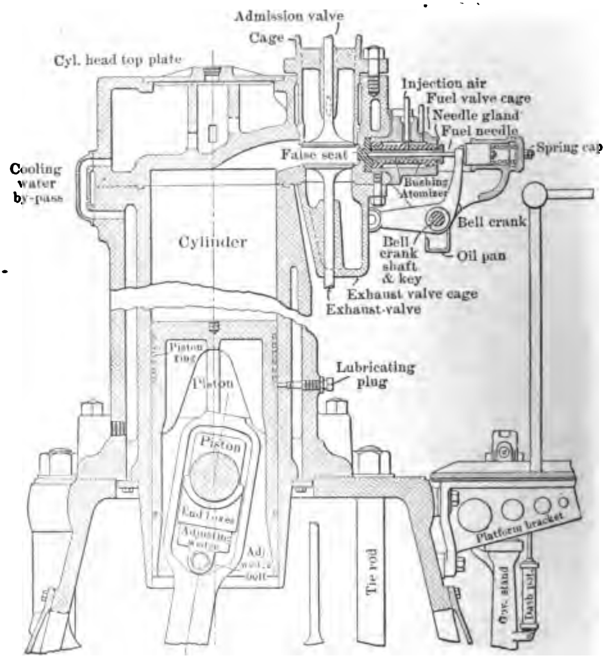


FIG. 134

THE BUSCH SULZER BROS. DIESEL ENGINE.—The four cycle Diesel engine manufactured by this company in St. Louis, Mo., is shown in section at Fig. 134.

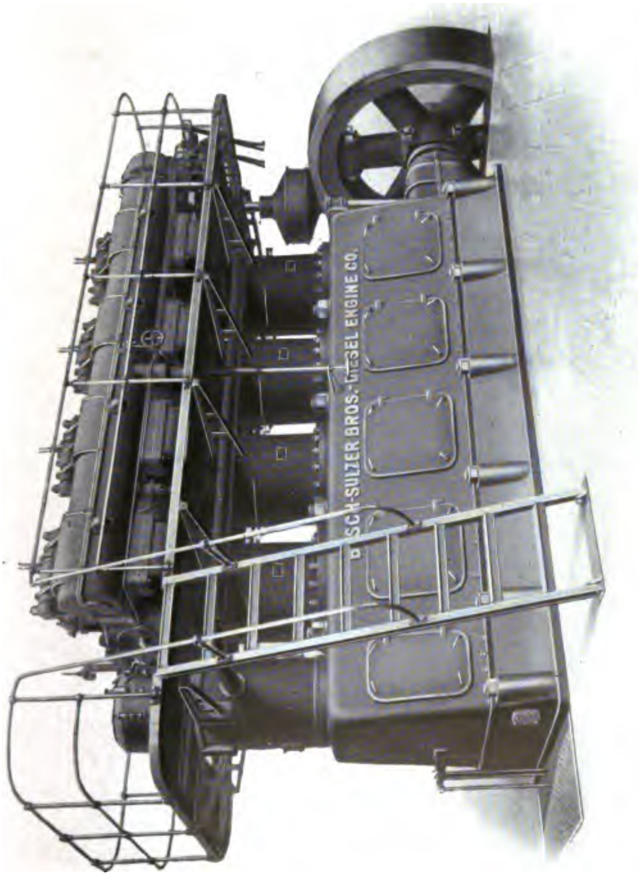


FIG. 135

This illustration shows the arrangement of the different valves, sprayer, etc., as hitherto built by this firm.

The later type of vertical four cycle four cylinder 500 H. P. Diesel engine now being built by this company is shown at Fig. 135. As will be seen from the illustration, the multi-stage air compressor for furnishing the injection air at about 1000 lbs. pressure is now operated directly from the main crankshaft by crank disc at the forward end. The crankcase is of the enclosed type reinforced with vertical tie rods—lubrication to all bearings is supplied by force feed pump. The oil inlet, air inlet, and exhaust valves placed in the cylinder head are operated by levers from the horizontal camshaft which revolves in enclosed oil case. The governor is mounted on the vertical shaft operated from the crankshaft which in turn is geared to the horizontal camshaft placed at the upper part of the cylinders.

Test on a Busch-Sulzer Bros.-Diesel Engine

The following test was conducted by A. C. Scott, Ph.D., on a 225 B.H.P. Diesel engine installed at Hugo, Okla., which was built by the Busch-Sulzer Bros.-Diesel Engine Company of St. Louis.

The engine was a regular four-cycle Diesel engine made by the above company which, at the time of the test, had been in operation for six months without adjustment. The only preparation made for the following test was examination of exhaust valves and checking over of valve settings to make sure they were properly timed. The tests extended over a period of

several days, the engine carrying its usual load, each test being made at such times as operating conditions permitted. The results, therefore, can be taken as representing ordinary operating conditions. The engine was direct connected to a 200 K.V.A. Fort Wayne 60 cycle 2300 volt generator, running at 164 R.P.M. Auxiliaries consisted of one belt driven three-stage air compressor operated by a 25 H.P. motor and a 10 K.W. exciter driven by belt from engine, the power for the compressor being furnished from an outside source. Net readings were taken of the power consumption of this motor, proper deduction being made from the K.W. output of generator. Fuel oil used was of the following characteristics, according to analysis of Doctors D. W. Harper and J. R. Bailey:

British Thermal Units	18,986. per lb. of oil
Specific Gravity (25.5° C. to 27° C.)	0.8531
Viscosity 33.3° C. (92° F.)	1.63
Flash Point	143.6° F.
Burning (Fire) Point	181.4° F.
Sulphur	0.2 per cent.
Water	Trace
Free Acid	None

All instruments used in the test were calibrated before using. The cooling water used was measured by meter readings, taken at 10 minute intervals, an average being taken for each three hour test. The electrical output was absorbed by a water rheostat.

TABLE V.

Number of Test	1	2	3	4	5	6
Date of Test	4–20–12	4–20–12	4–21–12	4–21–12	4–22–12	4–22–12
Duration of Test—Hours	3.00	3.00	3.00	3.00	3.00	3.00
K. W. Switchboard Reading	....	37.42	80.42	117.5	160	180.1
Comp. Motor, Gross K. W.	....	6.9	9.09	8.93	10	11.4
Net K. W. Output	....	30.52	71.14	108.57	150	168.7
Efficiency of Generator	....	85%	87%	90%	92%	92.5%
Net B. H. P. Load	2.25	49.7	111.39	162.97	219.63	245.6
Speed, R. P. M.	171.1	169.1	167.4	164.4	164.8	161.9
Pounds Cooling Water Per Hour	3187	3037	4875	6300	4537	5175
Jacket Inlet Temp. F°	61.8	62.4	61.9	62.6	61.7	66.7
Jacket Outlet Temp. F°	115.3	127.9	115.5	118.9	156.6	173.1
Injection Air Pressure—Atmospheres	39	45.6	56	59	66	77
Fuel Oil—Pounds Per Net K. W. Hour	....	1.25	0.7555	0.681	0.646	0.644
Fuel Oil—Pounds Per Net B. H. P. Hour	....	0.769	0.482	0.454	0.441	0.444
Fuel Oil—Gals. Per 100 Net B. H. P. Hour	....	10.8	6.8	6.4	6.2	6.2
Thermo-dynamic Efficiency (based upon net useful output)	....	17.4%	27.8%	29.5%	30.3%	30.2%

TABLE VI.

The analyses of the exhaust gases are given below:

	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6
Carbon Dioxide	1.50%	3.05%	3.92%	4.70%	6.46%	8.40%
Oxygen	18.42	16.03	14.95	14.70	9.93	8.93
"Illuminants"	0.00	0.00	0.00	0.00	0.00	0.00
Carbon Monoxide	0.00	0.00	0.00	0.00	0.00	0.00
Methane	0.00	0.00	0.00	0.00	0.00	0.00
Hydrogen	0.00	0.00	0.00	0.00	0.00	0.00
Nitrogen and Water Vapor	80.08	80.92	81.13	80.60	83.61	82.67
	100.00	100.00	100.00	100.00	100.00	100.00

The writer has given considerable attention to economical fuels available, among which is that of the crude oil produced in Mexico. The following is a table covering careful examinations of this fuel and mixtures thereof and gas oil, all made for the writer's special information.

TABLE VII.

	Spec. Gravity.	Baume Gravity 60° F.	Flash.	Fire.	Saybolt Viscosity.			Engler Viscosity.				Pour Test.	Sulphur.	Bud. T. Un.
					100° F.	130° F.	212° F.	20° C.	50° C.	70° C.	100° C.			
Mexican Crude....	.9626	15.5°	265	325	22.90	63.5	7.7	....	69.0	21.1	5.9	40	3.97%	18,257
Gas Oil	.8718	30.8	195	240	1.4	1.2	1.0	1.7	1.3	1.1	1.0	35	2.05%	19,092
90% Mexican Crude + 10% Gas Oil..	.9522	17.1	240	305	92.6	31.3	4.7	....	37.1	11.8	4.1	27	3.79%	18,340
80% Mexican Crude + 20% Gas Oil..	.9440	18.4	235	305	40.6	16.4	3.1	....	17.6	6.6	2.6	20	3.58%	18,424
70% Mexican Crude + 30% Gas Oil..	.9352	19.8	230	290	21.6	9.3	2.4	65.4	10.3	4.3	2.1	12	3.39%	18,507
60% Mexican Crude + 40% Gas Oil..	.9279	21.0	220	280	10.9	5.4	1.8	26.3	5.6	2.8	1.7	25	3.20%	18,591
50% Mexican Crude + 50% Gas Oil..	.9188	22.5	215	260	6.3	3.4	1.5	13.3	3.5	2.1	1.5	25	3.01%	18,674
40% Mexican Crude + 60% Gas Oil..	.9100	24.0	215	260	3.8	2.7	1.3	7.0	2.4	1.7	1.3	22	2.82%	18,758
30% Mexican Crude + 70% Gas Oil..	.8996	25.8	205	260	2.6	1.8	1.2	4.2	1.9	1.5	1.2	25	2.63%	18,841
20% Mexican Crude + 80% Gas Oil..	.8905	27.4	205	260	2.0	1.5	1.1	2.8	1.6	1.3	1.15	25	2.43%	18,925
10% Mexican Crude + 90% Gas Oil..	.8811	29.1	200	245	1.6	1.3	1.0	2.1	1.4	1.2	1.1	20	2.24%	19,008

TABLE VIII.

Ship.	Builder of Engine.	Cycle.	Cylinder.	Diameter.	Stroke.	Total H.P.	Year.	R. P. M.	Length.	Bdth.	Depth Mld.	Load Draft.	Speed in Knots.
Calgary.	A. B. Diesels, Motorer.	Two S. A.	Four each	11⅜"	16⅞"	520	1912	250	248'0"	42'6"	19'0"	14'0"	7½
California.	Burmeister & Wain.	Four.	Eight each	21¼"	28¾"	2200	1913	140	405'0"	54'0"	35'0"	23'3"	11¼
Christian X.	Burmeister & Wain.	Four S. A.	Eight each	20⅞"	28¾"	2100	1912	140	370'0"	53'0"	30'0"	23'6"	11½
Eavestone.	Richardsons, Westgarth & Co.	Two.	Four	20"	36"	800	1912	95	276'0"	40'6"	.....	....	10
Fionia.	Burmeister & Wain.	Four.	Eight each	29"	43"	*4000	1913	100	395'0"	53'0"	36'0"	24'3"	13½
Fordonian.	Clyde Ship Bldg. and Engine Co.	Two.	Four	15¾"	35⅜"	800	1912	140	257'0"	42'6"	16'10"	....	13
La France.	Schneider & Co., Carels	Two S. A.	Four	17¾"	22"	1800	1913	234	390'5"	57'3"	28'3"	23'8"	10
Arthur Von Gwynner.	Freriehs & Co. (Junkers)..	Two.	Four	17 5-16"	20½"	1700	1913	180	295'0"	44'0"	26'3"	21'4"	10
Hagen.	Krupp.	Two Twin Screw.	Six each	18⅞"	31½"	2200	1913	140	410'0"	53'0"	32'0"	26'0"	11
Juno.	Werkspoor	Two Single Screw.	Six	22"	39⅜"	1100	1912	125	258'0"	43'0"	19'10½"	18'6"	10½
Jutlandia.	Barclay, Curle & Co.	Four.	Eight each	20⅞"	28¾"	2100	1912	140	370'0"	53'0"	30'0"	23'6"	11½
Loki.	Krupp.	Two Twin Screw.	Six each	18⅞"	31½"	2200	1913	140	410'0"	53'0"	32'0"	26'6"	11

*I. H. P.

TABLE VIII—*Continued.*

Loudon.	Nederlandsche Fabrik.	Four.	Six	22"	39⅜"	1100	1913	135	278'0"	41'0"	19'0"	15'0"	10½
Emanuel Nobel.	Kolomnaer Mach. A. G.	Four Twin Screw.	Four each	19¼"	29"	1200	1909	150	380'0"	46'0"	25'0"	16'6"	10
Monte Penedo.	Sulzer.	Two Twin Screw.	Four each	18½"	26¾"	1700	1912	150	350'0"	50'0"	27'0"	....	10½
Pedro Christoffersen.	Burmeister & Wain.	Four Twin.	Eight each	19 11-16"	26"	1600	1913	140	362'0"	51'3"	25'6"	23'1"	10¾
Romagna.	Sulzer.	Two.	Foui each	12¼"	18⅛"	800	1911	250	175'5"	26'3"	12'7"	11'6"	12½
Rolandseck.	Carels.	Two.	Six	20"	36¼"	1500	1912	130	260'0"	40'0"	28'0"	18'0"	11½
Selandia.	Burmeister & Wain.	Four Twin Screw.	Eight each	20⅞"	28¾"	2100	1912	140	370'0"	53'0"	30'0"	23'6"	11½
Siam.	Burmeister & Wain.	Four Twin.	Eight each	23¼"	31½"	2400	1913	125	410'0"	55'0"	30'6"	26'5"	11¼
Suecia.	Burmeister & Wain.	Four Twin.	Eight each	22 1-16"	26"	1600	1912	140	362'0"	51'3"	25'6"	23'1"	10¾
Sebastion.	A. B. Diesels M. Co.	Two Twin Screw.	Six each	17¾"	21¼"	1600	1913	165	.....	....	.....	....	10½
Tynemount.	Mirrless Bickerton & Day.	Four Twin.	Four each	12"	13½"	600	1913	400	250'0"	42'6"	19'0"	14'0"	9
Wotan.	Carels Reiherstieg Co.	Two Single Screw.	Six	23⅝"	43"	*2300	1913	100	404'0"	52'6"	27'6"	23'0"	10
Vulcanus.	Werkspoor.	Four Single Screw.	Six	15¾"	23⅝"	500	1910	180	196'0"	37'9"	13'2½"	10'2"	8½
Toiler.	A. B. Diesels, Stockholm.	Two.	Four each	9⅞"	14½"	360	1910	250	248'0"	42'6"	19'0"	14'0"	6¾

The above table has been carefully compiled from various sources, including *Engineering, The Shipbuilder*, and the *Motor Ship and Motor Boat.*

*I. H. P.

INDEX.

APPENDIX I.

DIESEL ENGINE CASTINGS

BY

F. J. COOK

Being a paper read before the North East Coast Institution of Engineers and Shipbuilders, 30th January, 1920, at Newcastle-upon-Tyne, England, and reprinted by special permission of that Institution and the consent of the author, Mr. F. J. Cook.

DIESEL ENGINE CASTINGS

By F. J. COOK

[READ IN NEWCASTLE-UPON-TYNE, ENGLAND, ON FRIDAY, 30TH JANUARY, 1920.]

Monsieur Georges Carels, in the excellent address he gave to this Institution six years ago on Diesel engines, made the following significant remarks, which have a direct bearing upon to-night's subject.

"Success," he said, "is *not* ensured by securing good Diesel-engine designers success being *far more* a question of manufacture and *metallurgy* than some people suspect, and experience in the right material to use is essential."

"The problem becomes more a question of *metallurgy* than design and it is only by close collaboration between the foundry and the designing office, and a good deal of experimenting, that the *right* solution will be found."

"The enormous quantities of heat transmitted cause very high stresses in the castings, and only special sorts of cast-iron can withstand such stresses. It is highly to be desired that more extensive study be given to the question of appropriate cast-irons for internal combustion engines. On the other hand such intricate castings as a cylinder cover should be suitably heat-treated to relieve them of any *casting stresses.*"

The italics are the Author's but the fact remains that not only Monsieur Carels, but every one who has had to do directly with the manufacture of Diesel engines, is convinced that the metallurgical side of the question is one of vital importance. This being the case, it is remarkable that among the whole host of papers which have been written with regard to this type of prime mover the Author has only been able to find two which have made any attempt to deal with the castings problem.

The Author therefore proposes to deal only slightly with design, and that only as far as it affects the metallurgy of the subject, and to deal more fully with the items which count in the production of successful castings.

Messrs. Alexander Outerbridge,* H. F. Rugan and H. C. Carpenter,† Dr. W. H. Hatfield‡ and others have made a particular study of the phenomena of growth of cast-iron after repeated heating.

The results of their investigations are so well known that it is unnecessary to deal extensively with this part of the subject. Suffice it to point out that they have proved that growth is largely due to the influence of gases which penetrate the cast-iron by way of cavities formed by large graphite flakes or loosely intermixed crystals of the metal. Therefore anything that favours either of these two undesirable conditions, whether a composition of the metal which will allow the formation of large graphite plates or a design which

* Outerbridge. Franklin Institute, January, 1904.
† Rugan & Carpenter. *Journal* Iron & Steel Institute, May, 1909.
‡ Hatfield. *Journal* Iron & Steel Institute. May, 1907.

tends to a loose intermixture of the crystals and low intercrystalline cohesion, is of necessity bad.

CRYSTALLIZATION.—In cast-iron when it is cooling down from the molten to the solid state, the planes of

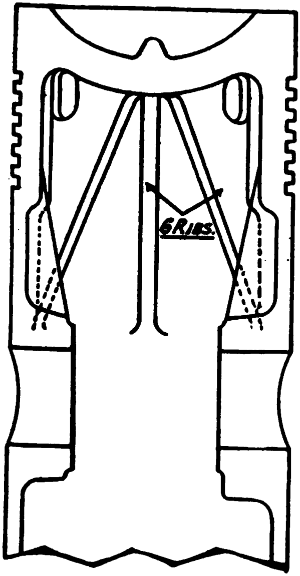


FIG. 1

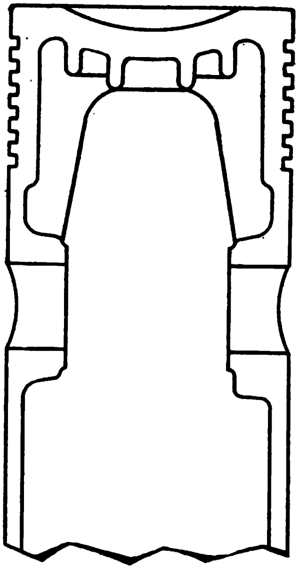


FIG. 2

crystallization group themselves perpendicularly to the surface of the external contour, that is to say, at right angles to the outside surface and in the direction in which the heat of the fluid cast-iron has passed outward from the body in cooling and solidifying.

Every abrupt variation in the external contour, every salient and every re-entering angle, no matter how small, upon the surface of a casting, is attended with

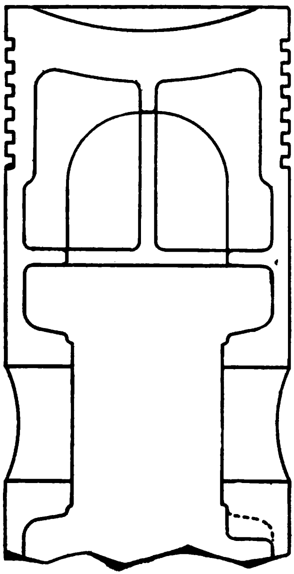


FIG. 3

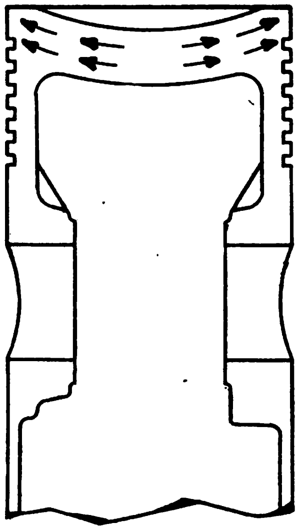


FIG. 4

an equally marked sudden alteration in the arrangement of the crystals of the metal. In the neighborhood of each such point of variation there occurs a confused and irregular formation of crystallization, so that the proper interlocking of the crystals is prevented and

lines of weak and loosely packed crystals, if not actual cavities, are produced.

There are castings of this character which, on ac-

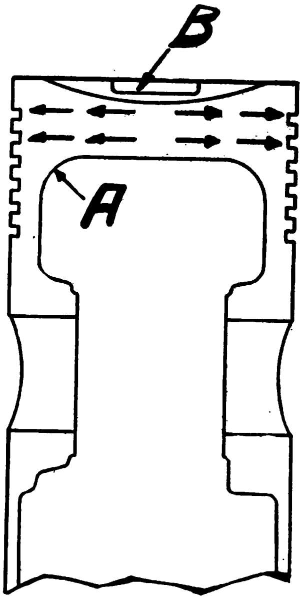


FIG. 5

count of the static pressure of the metal in the mould when casting, or some other cause, artificial or accidental, do not show these weaknesses in any marked degree; but when such castings are exposed to repeated heating the crystals change position and form the same

loose and confused mass, with the consequent weakening of the structure.

The natural remedy for this, therefore, is to avoid all sharp angles, ribs, and sudden deviations of sectional outline, so that the metal while being poured may be allowed to flow in natural curved lines. These remarks are equally true of castings in general, but are of primary importance in the design of Diesel engine pistons and cylinder heads.

Figs. 1, 2, and 3 represent designs of Diesel engine pistons which have been culled from the illustrations of these parts appearing in technical journals. In the light of what has been previously said, it is apparent that the castings, on account of the ribs, are not only subject to the effects of bad crystallization mentioned, but also have the disadvantage of being subjected to high initial cooling strains which further increase the liability to cracking when in use.

Fig. 4 shows a much improved design, there being quite an absence of anything that will retard a natural formation and an even cooling down in the casting. This design has been quite successful in ordinary work. By designing the under side of the head straight, as in Fig. 5, a further improvement has been effected. Such a formation allows a more easy passage for dissipation of the heat to the outside cooling surface provided for by the water-cooled liner, as indicated by the arrows. This type has given excellent results during the war in submarine work.

It is essential that there be a good easy angle at the point marked A in order to avoid a too abrupt change

of section. Also, an angle at the bottom of the cutaway B (where such is required to miss the valves in the cylinder cover) is most desirable, for it must be remembered that a definite sudden change in contour brought about by a sharp corner forms a point from which a crack is very likely to start.

Every advantage should be taken of any process in manufacture that will facilitate closeness of grain with its consequent close packing and greater cohesion of the crystals. It is therefore advantageous to cast the cylinder liners and pistons on end, with the breach end at the bottom, thus taking advantage of the static pressure of the metal in the mould. Denseners also have been used for improving the compactness of the piston head, but the use of these requires special care or hardness will be overdone.

Hardness is a point which requires very careful watching in this class of work, as a cast-iron piston if too hard, has a tendency to split right across when first put to work, particularly if it is subjected to full load soon after starting. A tough iron therefore is more desirable.

The properties of cast-iron which are essential to successful working of Diesel engine pistons and liners are: High tensile and other physical properties; metal which will readily take a high polish under working conditions and thus give good resistance to wear, and the ability to resist growth and cracking when subjected to the high working temperatures which are a feature of this class of engine. These conditions are only met with in cast-iron possessing the highest intercrystalline

cohesion, and this is also associated with metal which gives the highest tensile strength. It is a remarkable and fortunate circumstance that all these desirable qualities are associated with cast-iron in proportion to its tensile strength. This naturally narrows the choice, and materially assists in determining the proper metal to be used.

The factors which tend to give a cast-iron having the highest physical properties are:

(1) Chemical composition.
(2) Casting temperature.
(3) Rate of cooling.
(4) Microstructure.

It is essential to take into account the rate of cooling when considering the chemical composition of metal to be employed in any given casting. For the purposes of this Paper, therefore, the proportions of the various chemical elements suggested are based upon those which will give the highest tensile strength on a bar cast 1¼ inches diameter parallel and turned down to ½ a square inch or thereabouts in the centre.

CARBON.—The most important element in cast-iron is carbon. It is not too much to say that all the physical properties of the metal depend upon the quantity of this element present and its condition, the value of the other elements being in accordance with the effect they have on the carbon and the compounds it forms. It is very seldom that one finds any *particular* mention of the *total* quantity of carbon present, whilst the quantity of combined carbon advisable is

quite glibly spoken of. Yet it must be obvious that with the same percentage of combined carbon present there must be a great difference in the condition of the metal if the total carbon varies, say from 3 to 3.75 per cent., particularly when Dr. Stead* has shown examples where an increase of each 0.1 per cent. of graphite has reduced transverse strength of 2 cwts. and tensile strength by 0.8 tons per square inch.

Professor Turner, in his admirable book, "The Metallurgy of Iron" (p. 251), states that maximum tensile strength is associated with 0.47 per cent. of combined carbon and maximum transverse strength with 0.7 per cent. The Author's experience, however, leads him to the conclusion that these figures for present day practice should be reversed, maximum tensile strength being obtained with 0.6 to 0.8 per cent. and maximum transverse strength with 0.4 to 0.6 per cent. and with total carbon not exceeding 3.25 per cent.

SILICON.—One of the most important elements in determining the suitability of cast-iron for any given purpose is silicon. This element is always present, in proportions most variable, whilst the influence it exerts on the condition of the carbon present, and consequently on the hardness and fluidity of the metal, is most marked. Professor Turner's research on the effects of this element on cast-iron, though made some 30 years or so ago, has stood the test of time, and probably it is not too much to say, laid the foundation for the application of the science of chemistry to practical working in cast-iron.

* British Foundrymen's Association *Proceedings*, 1915.

As the quantity and condition of the carbon have been stated to be so important, whilst the effect of silicon on this element is so marked, it is clear that if a formula can be adduced that will show when these two elements are in the best proportions for any given class of work, a very important stage will have been reached. With this object in view, the Author has adopted the following more or less empirical formula, based upon saturation point of carbon, and he has used it with marked success in practical working:—

$$Sc = \frac{C}{4.26 - \frac{Sil}{3.6}}$$

Where Sc = ratio of silicon to carbon
C = *total* carbon in percentage.
Sil = silicon in percentage.

The elements carbon and silicon are in the best proportion for Diesel-engine pistons and liners and all parts requiring highest tensile strength when the value of Sc = 0.76 to 0.82, and for water-cooled cylinder heads and castings requiring maximum transverse strength when the value of Sc = 0.83.

SULPHUR.—By many, sulphur is looked upon as the arch-enemy of the ironfounder; but although its presence makes the molten metal thick and sluggish, thus giving rise to blow-holes, this element, in fair proportions—say, 0.12 per cent.—adds considerably to the wearing properties and strength of the metal. In every

case, however, sulphury iron *must* be cast as hot as possible if trouble is to be avoided.

PHOSPHORUS.—There are many who think it impossible to produce strong irons without the percentage of phosphorus being very low. To lower this element they frequently resort to hematites, but these, with their high percentage of total carbon, often do more harm than a liberal amount of phosphorus would. When the proportions of total carbon and silicon are those required for maximum tensile strength—as in the formula given earlier in this Paper—the proportion of phosphorus may be as high as 1 per cent. without seriously jeopardizing the tensile strength; in fact, with steam cylinders and castings requiring maximum strength and of complicated design the reduction of shrinkage and the additional fluidity of the metal due to this percentage will generally result in a sounder casting, with fewer initial contraction strains. Where the ratio of silicon and carbon is different from that which has been suggested, the quantity of phosphorus becomes one of the vital elements and must then be kept as low as possible.

In the early part of this Paper reference was made to the fact that only two investigators were known to the Author to have dealt with this particular problem. One of these, Mr. J. Edgar Hurst, has contributed two excellent Papers* in which he has dealt in an exhaustive manner with many of the troubles to be

* Manchester (England) Association of Mechanical Engineers *Proceedings,* 1916.
Institution of British Foundrymen's *Proceedings,* 1917-18.

met with in cast-iron in the working of Diesel engines. One of the conclusions he has arrived at is that a high percentage of phosphorus is detrimental in this class of work.

Unfortunately the chemical analyses and physical properties given by Mr. Hurst do not fulfil the requirements of the formula which has been stated and thus, the composition having started from a poor basis, the quantity of phosphorus has become of critical importance.

TABLE I.

	Suggested by Hurst.	German.	Another German.	Continental	Replaced by	Scotch.	Suggested by the Author
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Phos. ..	0.52	0.33	0.59	0.54	1.34	0.54	not over 1
Mang. ..	1.44	0.72	0.99	1.08	0.28	0.97	not over .5
Sc. ..	0.9	0.83	0.91	0.92	0.8	0.77	0.76–0.82
Results ..	—	good	bad	bad	good	good	has never given star cracks, low tensile results, or bad wearing qualities.

Table I., which contains part of the information gleaned by the Author from several years' study of British and Continental results, goes to show that where the silicon and carbon are in the desired ratios, the quantity of phosphorus present does not become so extremely important.

The second column gives particulars of a piston in a German-made engine. This engine has given reasonably good results and is still at work. After about six months running the pistons were found to be slightly deformed, and showed signs of wear, but they would generally be classed as good. In the engine of German make dealt with in the third column, the piston did not last six months, having worn badly and star-cracked, but certainly not because the phosphorus was not low enough or the manganese not sufficiently high. With the next engine, described as "Continental" the results in working were bad from the first. Without any alteration of design, and specially to prove the effects of phosphorus, the replace pistons were made with this element as high as 1.34 per cent. but with a silico-carbon ratio to satisfy the formula, whilst the manganese was very low, being only 0.28 per cent. These pistons are still giving excellent results and the engineer's first report was that they ran better than the original ones had ever done. The engines of the type dealt with in the sixth column and described as "Scotch" have also given excellent results. Many pistons of this class of metal were employed in submarine work during the war. The last column contains the corresponding particulars recommended by the Author.

In each of the instances cited—and many more could be given if necessary—the results in actual working bore no relation to either the phosphorus or the manganese content of the metal, but were closely

related to the results suggested by the silico-carbon ratio.

MANGANESE has a tendency to harden the iron both directly and by causing the carbon to remain in the combined form. In the presence of much sulphur, however, it may act as a softener by uniting with that element and forming manganese sulphide, which rises to the top of the metal and passes away with the slag. When adding the manganese to the metal, in the form of ferro-manganese, in the ladle it is necessary that the metal be very hot and that sufficient time be allowed for the change to take place, otherwise the manganese-sulphide will be found in the top part of the casting in the form of very hard spots which are difficult, if not impossible, to tool by ordinary methods. The general properties of the metal—other than a tendency to chill—are not materially affected so long as the manganese is not in greater proportion than 0.7 per cent. For castings in the making of which denseners are employed on faces which have to be machined, it is not advisable to have more than 0.4 per cent.; otherwise there will be a tendency to chilling.

As manganese tends to the removal of sulphur and the formation of brittle spikey crystals, which break off under rubbing, much of this element is not good for castings which are required to have good wearing properties when working under heated conditions.

To sum up, the proportions of the various chemical elements which the Author has found to give good results for Diesel engine cylinder liners and pistons are:

Total carbon	=	3 to 3.2 per cent.
Silicon	=	1 to 1.2 per cent.
Phos.	=	not over 1 per cent.
Sulphur	=	0.12 per cent.
Manganese	=	not over 0.5 per cent.

For cylinder covers, which require to be not quite so hard for wearing purposes, and in which metal with rather less contraction is desired, the silicon can with advantage be increased to 1.5 per cent and the manganese to 1 per cent.

After twenty years' experience with this class of metal the Author finds that the most advantageous amount of steel to be added to the mixture is 15 per cent. Experiments with varying amounts up to 40 per cent have been made, but with above 15 per cent the metal has a tendency to be too hard, which is very undesirable for this class of work. Further improvement will probably be gained by the addition of some of the rarer elements and in this direction Vanadium-Chrome or Nickel-Chrome seems to hold out some prospect of success.

Whilst chemical analysis gives the quantities of the elements present, the quality of the metal depends also upon how they exist as compounds and upon their association and relative distribution through the whole mass—just as in the case of the human body, it is not only essential to have the *correct proportions* of bone, muscle, etc., but all must also be related to each other correctly if the person is to be healthy and strong. It is therefore essential that the etched surface of the metal

as seen under the microscope shall consist of large areas of pearlite and moderately small areas of cementite and phosphide eutectic, with the graphite in a finely divided form. Such a structure is illustrated in Fig. 6.

SURFACE WEAR.—There is much room for investigation into the subject of surface disintegration and wear of cylinders and all rubbing surfaces of cast-iron. When cast-iron is machined by the ordinary cutting methods there is more or less tearing away of the crystals. The tearing out of the softer material produces cracks or indentations and leaves the harder constituents standing up in relief. The degree or fineness of the finish will depend upon the structure of the material and the speed at which the material was cut; generally speaking, according to Professor Poliakoff* the quality of the finish will be improved as the cutting speed decreases. The resistance to wear under working conditions depends upon how far the crystals will resist breaking off—thus forming a hard, gritty abrasive substance which increases the wear—or will spread over and form a highly polished surface. The existence of large pearlite areas shown in Fig. 6, with the small areas of harder constituents and fine graphite, has a marked effect in bringing about the latter condition.

Grinding of cylinder liners and piston rings, being less drastic and exposing less of the harder crystals, appears to be a desirable method of finishing these articles, provided every trace of emery be removed.

* Manchester (England) Association of Engineers, 1915.

Where this is not practicable, very slow cutting speed and fine feeds for finishing should be resorted to with the object of decreasing the wear.

Another essential feature of the structure is that the cementite and phosphide eutectic shall have taken up a well defined network structure, in accordance with the

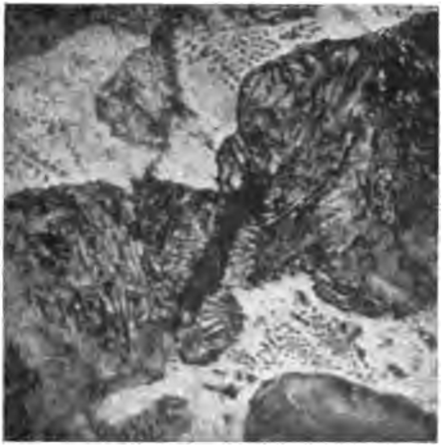


FIG. 6

research carried out by the late George Hailstone and the Author.† A good example of this is shown in Fig. 7.

ANNEALING.—Considering the enormous quantities of heat which the pistons and cylinder heads have to deal with, it is imperative that everything in the nature of cooling strains in the castings should be eliminated,

† British Foundrymen's Association, *Proceedings,* 1908-9.

so that the crystallization shall be in its natural formation, allowing free and ample expansion. As all castings are more or less subject to initial strains, it is desirable that those having to work under heated conditions should undergo some heat treatment or

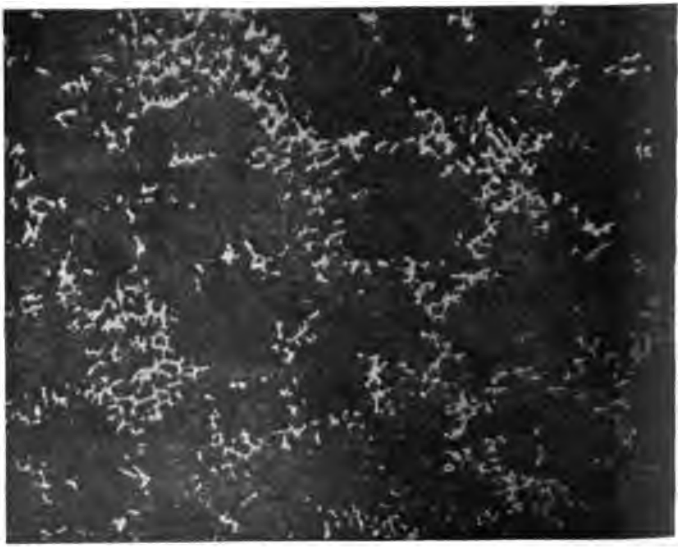


FIG. 7

annealing. It is to be feared, however, that frequently more harm than good is done owing to a general lack of knowledge of what is desired and of the results generally accruing from heat treatment of cast-iron, and to the crude manner in which the heat treatment is performed.

Dr. Hatfield* has carried out quite a large number of

* Hatfield, Iron & Steel Institute *Proceedings,* 1907.

experiments showing the effects of heat treatment of cast-iron which are well worth the attention of those who have to do with Diesel-engine castings. Whilst heat treatment of cast-iron, even at a low temperature, affects the physical properties of the metal, a prolonged treatment at 780°C. is positively dangerous, as, among other things, the pearlite is then broken down and the generally desirable physical properties of the metal are destroyed.

Table II. gives the results of annealing on the physical properties of a few out of many test bars of cast-iron of the composition recommended by the Author. These tests have been carried out by Mr. J. J. Howell, of Birmingham. The heat treatment to which the castings have been subjected consists of gradually raising the temperature to 600°C., retaining that heat over six hours and cooling down in the furnace over a period of 48 hours.

TABLE II.

Test No.		Tensile Strength. Tons per Square inch.	Transverse Strength. Cwts.1′ sq. bar 12″ centre.	Impact.	Hardness.
1	As cast ...	17.0	35.5	65	77
	Annealed ...	13.7	32.0	60	44
2	As cast ...	16.9	31.25	75	84
	Annealed ...	13.5	27.5	50	39
3	As cast ...	17.8	33.75	70	80
	Annealed ...	13.77	28.25	60	67

The tensile test bars were cast $1\frac{1}{4}''$ diameter, turned down after annealing to $\frac{1}{2}$ square inch area in centre before testing. The transverse bars were cast $1\frac{1}{4}''$ square, machined to $1''$ square, and tested on $12''$

centres. The impact bars and tests were in accordance with the notes on impact given later in this Paper. The hardness numeral was obtained by the drill method.

It will be observed that in every case the physical properties have deteriorated. The same heat treatment continued for half the length of time would in all probability be sufficient to remove the casting strains in cylinder heads and pistons, with a corresponding decrease in the deterioration of the physical properties. Considering the cooling effects the cylinder liner receives when at work, and bearing in mind that it is less subject to initial casting strains, the Author does not consider heat treatment of these parts necessary.

It is essential that the heat treatment shall be carried out in a properly designed furnace. Owing to the castings to be treated being generally too large to allow of their being packed in pans and surrounded with suitable packing they should only be subjected to radiated heat. If they are exposed to the action of a coal-fire flame, deterioration of a serious nature will take place, as shown in Table III. The composition of metal as cast was as recommended for cylinder heads, temperature and time of heat treatment, size of bars, and method of testing being the same as in Table II.

TABLE III.

Test No.		Tensile Strength. Tons per Square inch.	Transverse Strength. Cwts.1′ sq. bar 12″ centre.	Impact.	Hardness.
4	As cast ...	17.3	34.75	75	82
	Annealed ...	15.49	33.50	55	97
5	As cast ...	17.8	32.0	70	78
	Annealed ...	14.4	30.75	60	110

The most notable change shown is the increase in hardness, which has previously been pointed out as being a very important condition.

The alterations in chemical composition of the parts of the bars subjected to the action of the flame were:

Total carbon decrease from 3.163% to 3.027%
Sulphur increased from .. 0.112% to 0.17%
Silicon decreased from ... 1.586% to 1.306%

Tests

Two useful workshop tests—with the class of iron suggested—for the guidance of the founders are: Impact test and casting temperature.

Impact Test.—Whilst impact testing is not usually applied to cast-iron, it is, however, beginning to be used, particularly on the Continent, in connection with Diesel-engine work. Whilst the Izod method of testing is more applicable to brittle substances, and should therefore be of use for cast-iron, the practical limits as to the size of the bar are too low to give correct indications for this class of work. The sections of Diesel-engine parts, particularly the piston heads, are very thick, and it is more advisable to test the metal under nearly similar conditions.

A satisfactory test is on bars 40 mm. square (1.57 in.) supported on knife edges 160 mm. (6.3 in.) apart, and by dropping a weight of 12 kilos (26.5 lbs.) from a height of 30 cm. (11.8 in.), increasing the height of drop by increments of 5 cm. (1.9 in.) until the sample breaks, the height of drop at which the bar eventually

fractures being taken as the impact figure. Attached to the weight, in such a manner as to strike the bar parallel to the supporting knife edges, and at the centre, is another knife edge. The face of each knife edge should be rounded $\frac{1}{16}$ in. radius.

A result of 55 cm. (21.5 in.) is considered none too high for this class of work, although this is quite a severe test. The maximum attained, as far as the Author knows, has been 80 cm. (31.3in.). Some recent results obtained with iron conforming to the suggested conditions are shown in Table II.

CASTING TEMPERATURE.—For many years foundrymen have been well conversant with the fact that casting temperature has a marked effect upon the physical properties of cast metals, but it was not until the important research of Dr. Longmuir* that anything like exact results were obtained. Many of those results were at the time certainly startling. In connection with the casting of high-class metals of the cast-iron series there is undoubtedly a great need for a handy practical optical pyrometer. For small quantities of metal the "Foster" pyrometer is very convenient, but for large quantities of metal it is of very little use.

Low silicon and low total carbon cast-iron are very susceptible to the effects of casting temperature, and some irons employed for such purposes are also liable to liquid contraction. A handy workshop test for casting temperature with this class of iron, one which the Author has used successfully for many years, con-

* Iron & Steel Institute; Carnegie Memoirs.

sists in making bars of the general dimensions shown in Fig. 8. When cold, the bars are broken through the line A.B., the condition of the fracture giving an indication of the temperature at which the metal was

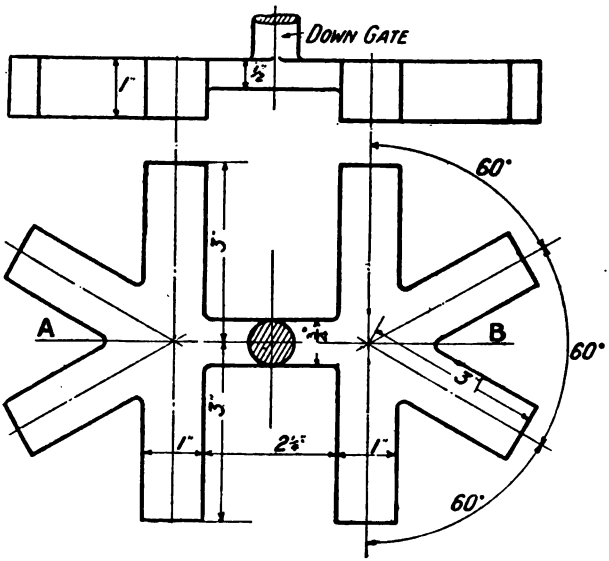


FIG. 8

poured. Passing from the correct temperature (which is usually as hot as can be obtained from the cupola) to a very low one, the following fractures will be observed. Perfectly solid and homogeneous; then slight whitish centre; then brown centres of vary-

ing sizes and depth of colour, as temperature is lowered, and then black centre. With the latter are always associated open cavities, and immediately above each cavity is invariably found a gas hole. A set of results showing various casting temperatures is shown in Fig. 9.

FIG. 9

The Author desired to make the Paper as comprehensive as possible. In order to cover all the ground within a reasonable space he has found it necessary to state many things tersely, and in some cases with what may have seemed dogmatic abruptness; but if

all the arguments had been given in full and the experiences which had led up to many of his conclusions had been detailed, the Paper would have run to the dimensions of a book. The Author hopes, however, that he has provided sufficient material to ensure a full and free discussion of a subject which is of the highest importance to the engineering and foundry trades.

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BY

MICH. E. TOEPEL

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